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AUDELS ENGINEERS MECHANICS GUIDE 4

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MODERN ENGINEERING PRACTICE

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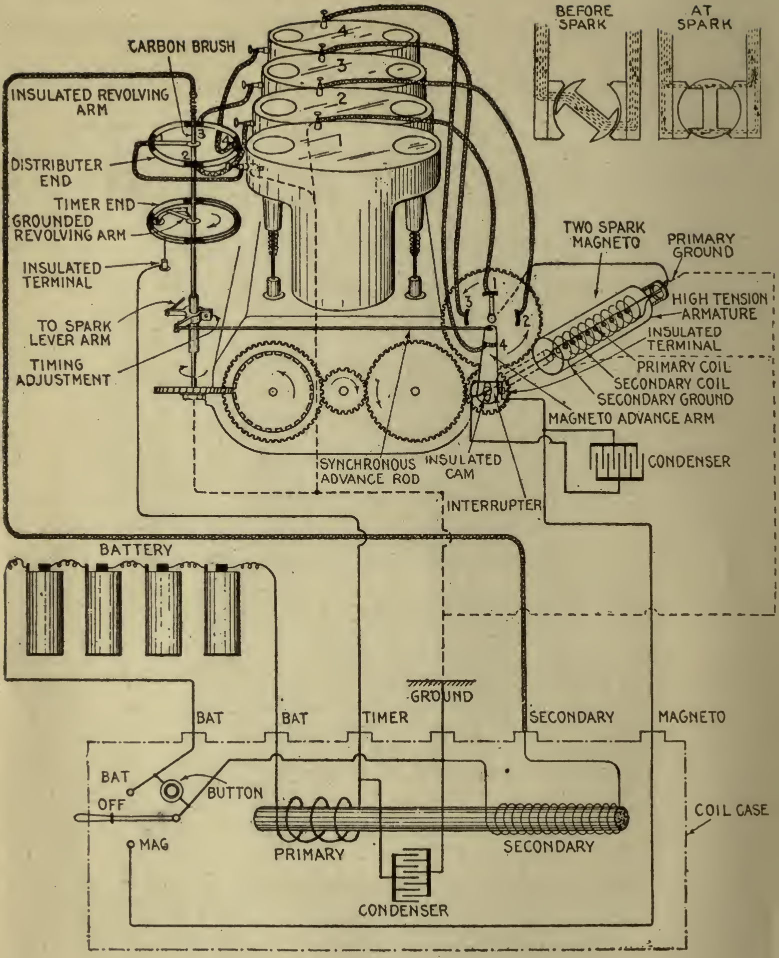

Plate 4.—Elementary Double Ignition System for Four Cylinder Automobile Engine.

It consists of a two-spark high tension magneto system and a battery synchronous ignition system with engine driven distributer.

AUDELS ENGINEERS AND MECHANICS GUIDE 4

A PROGRESSIVE ILLUSTRATED SERIES
WITH QUESTIONS-ANSWERS
CALCULATIONS

COVERING

MODERN ENGINEERING PRACTICE

SPECIALLY PREPARED FOR ALL ENGINEERS
ALL MECHANICS AND ALL ELECTRICIANS.
A PRACTICAL COURSE OF STUDY AND
REFERENCE FOR ALL STUDENTS AND
WORKERS IN EVERY BRANCH OF THE
ENGINEERING PROFESSION

BY

FRANK D. GRAHAM, B.S., M.S., M.E.

GRADUATE PRINCETON UNIVERSITY
AND STEVENS INSTITUTE-LICENSED
STATIONARY AND MARINE ENGINEER

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NOTE

In planning this helpful series of Educators, it has been the aim of the author and publishers to present step by step a *logical plan of study in* ***General Engineering Practice,*** taking the middle ground in making the information readily available and showing by text, illustration, question and answer, and calculation, the theories, fundamentals and modern applications, including construction *in an* **interesting** *and* **easily understandable** *form.*

Where the question and answer form is used, the plan has been to give *short*, *simple* and *direct answers*, limited to one paragraph, thus simplifying the more complex matter.

In order to have adequate space for the presentation of the important matter and not to divert the attention of the reader, descriptions of machines have been excluded from the main text, being printed in smaller type under the illustrations.

Leonardo Da Vinci once said:

> "Those who give themselves to ready and rapid practice before they have learned the theory, resemble sailors who go to sea in a vessel without a rudder"

—in other words, "*a little knowledge is a dangerous thing.*" Accordingly the author has endeavored to give ***as much*** *information* ***as possible*** in the space allotted to each subject.

> The author is indebted to the various manufacturers for their co-operation in furnishing cuts and information relating to their products.

These books will speak for themselves and will find their place in the great field of Engineering.

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CHAPTER 46

GAS ENGINES

The term gas engine is generally understood to mean an internal combustion engine using natural gas or the vapor of gasoline for fuel as distinguished from "oil" engines or those using kerosene or heavy low grade oils, the latter class including the Diesel engine.

The gas engine is a form of heat engine, since it operates by means of the expansive energy of a gas which has been subjected to the action of heat. There are two kinds of heat engines:

1. External combustion engines;
2. Internal combustion engines.

In the first type the combustion of fuel takes place outside the working cylinder as in steam engines or hot air engines, while in the second type combustion takes place inside the working cylinder as in gas and oil engines.

It is a matter of ordinary observation that heat, by expanding bodies, is a source of mechanical energy, and conversely, that mechanical energy being expended either in compressing bodies, or in friction, is a source of heat; these truths are summed up by the first law of thermodynamics,* as follows:

Heat and mechanical energy are mutually convertible in the ratio of about 778 *foot pounds for the British thermal unit* (Wood).

---

*NOTE.—Thermodynamics, the science of heat considered as a form of energy, is useful in advanced study of the theory of gas, steam, and air engines, refrigerating machines, compressed air, etc. The method of treatment usually adopted is severely mathematical, involving constant application of the calculus. The subject will not be presented here further than the discussion of a few fundamental truths which should be understood by those interested in engines of any kind.

Although heat and work are mutually convertible and in definite and invariable proportions, yet no conceivable heat engine is able to convert all the heat given to it into work. To get a continuous supply of mechanical energy from heat depends upon getting a continuous supply of hot and cold substances; it is by the alternate expansion and contraction of some substance, as a gas, that heat is converted into mechanical energy.

Perfect heat engines are ideal conceptions of machines which are practically impossible, but whose operations are so arranged that, if possible, they would convert the greatest conceivable portion of the heat given to them into mechanical work. The efficiency of a heat engine is the ratio of the heat converted into mechanical work to the total amount of heat which enters the engine.

The efficiency of a perfect heat engine depends on two things: 1, the temperature of the heat received, and 2, the temperature of the heat rejected. The greater the difference between these temperatures the greater the efficiency, that is, the greater will be the proportion of the total heat converted into mechanical energy, and the smaller the proportion of the total heat which necessarily passes by conduction from the hot to the cold body.

The second law of thermodynamics is an expression for the efficiency of a perfect elementary engine (Wood), and is stated as follows:

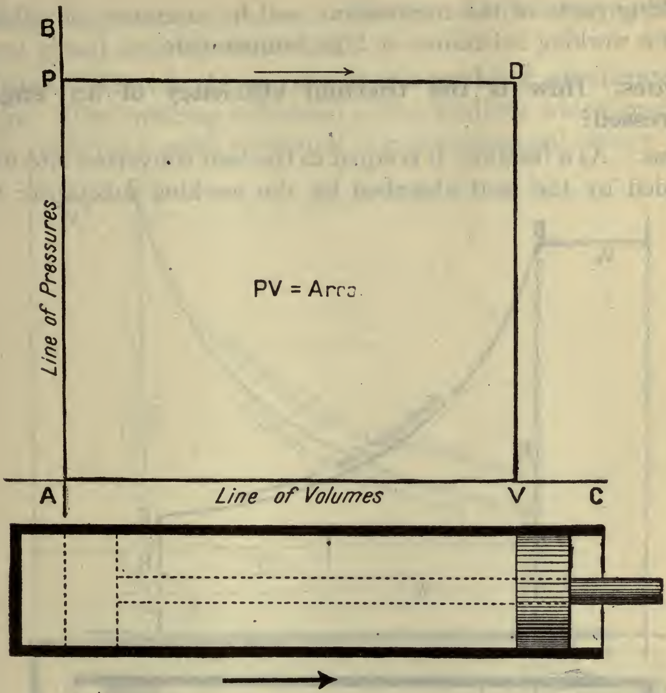
It is impossible for a self-acting machine, unaided by any external agency, to convert heat from one body to another at a higher temperature. (Clausius).

Ques. What is the practical interpretation of the two laws of thermodynamics?

Ans. According to the first law, there is nothing to prevent an engine converting all of the heat supplied to it into mechanical energy, and its efficiency may be expressed by 1 or unity. According to the second law, no heat engine can convert into mechanical energy more than a small fraction of the heat supplied to it, the greater portion being necessarily lost in various ways.

Ques. What causes these heat losses?

Ans. In overcoming the molecular resistance of the working substances to changes of form; in overcoming the friction of the

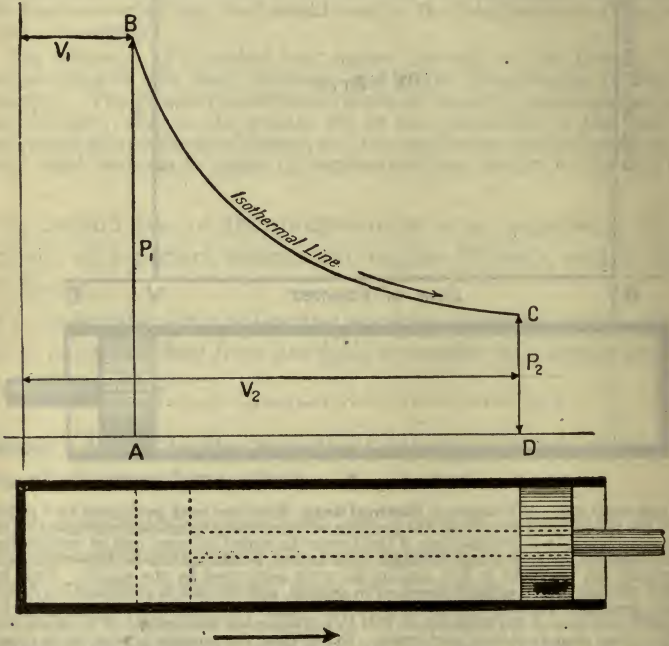


FIGS. 2,638 and 2,639.—P V diagram; ***thermal lines.*** Since the work performed by a piston engine is the product of two factors, the pressure of the working substance multiplied by its increase in volume, it is obvious that if the former be stated in some unit of weight such as pounds, and the latter in a unit of length such as feet, a diagram can be drawn enclosing an area which will represent in foot pounds the work performed by the engine. Thus in the figure if the line A B, represent pressures in pounds, and the line A C, volumes in feet, then for any pressure P, required to move a piston from A, to a distance corresponding to the volume V, the area of the rectangle A P D V, will represent the amount of work performed, or P V = foot pounds of work performed. In this case, the pressure is assumed as constant, and the example illustrates the simplest form of a P V-diagram. Usually, however, the pressure is not constant, but decreases as the volume increases, therefore the line P D, will assume the form of curve sloping downward from P to D, the varying ordinates of which will represent the pressures for the corresponding volumes, and the area enclosed the amount of work performed. In this case, the temperature is assumed as constant, and the curve represents the expansive action of a gas under that particular condition. The action of a heat engine is modified by the effects on its working substance of other conditions of temperature, volume, and pressure, which obtain at the time heat is applied. The graphical representations of the thermal lines resulting from those conditions will give areas which will differ, as later shown, for the various classes of heat engine, according to the method of heat application, and also according to the practical working cycle employed for the various types of any particular class.

working parts of the mechanism, and by rejection, or exhaust of the working substance at high temperature.

Ques. How is the thermal efficiency of an engine expressed?

Ans. As a fraction; it is equal to the heat converted into work divided by the heat absorbed by the working substance; this

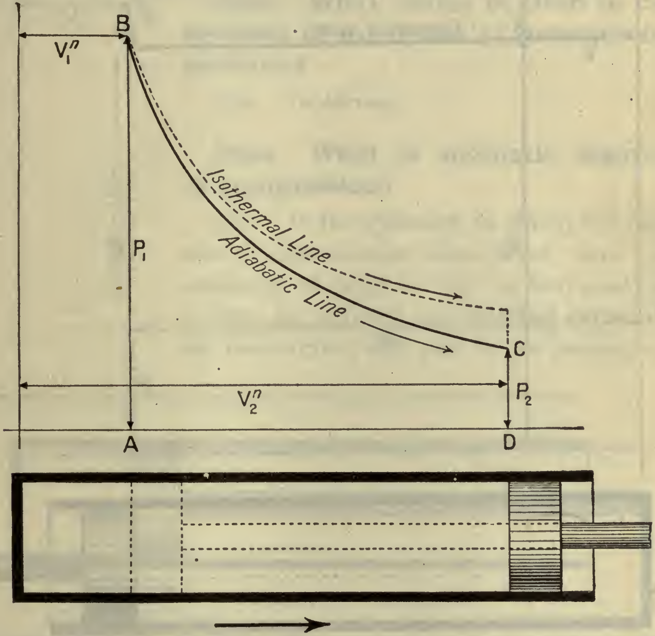


FIGS. 2,640 and 2,641.—***Isothermal line.*** The most natural conditions for the expansion or compression of a gas doing work in the cylinder of an engine may be stated as follows: 1. For expansion, when the pressure decreases as the volume increases, with a corresponding decrease in temperature due to the external work done by the gas in moving the piston. 2. For compression, when the pressure increases as the volume decreases, with an increase in the temperature due to the work spent upon the gas during the act of compression. If, however, suitable means be provided, so that during expansion the gas absorbs an amount of heat equal to that expended in moving the piston, and during compression it rejects an amount of heat equal to that which is spent upon it by the act of compression, or in other words, *if the* ***temperature*** *of the gas be kept* ***constant*** *its mode of expansion or compression will be* ***isothermal,*** or the temperature of the gas will be the same at all points of the piston stroke, and the volume at any point multiplied by the pressure will be the same or $PV = \text{Constant}$.

fraction represents the thermal efficiency of an engine operating under actual working conditions.

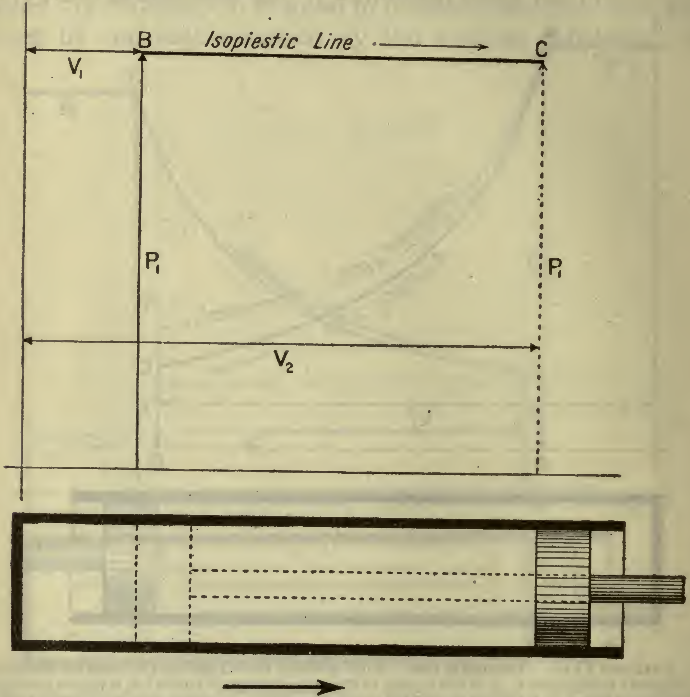
Ques. What is the nature of the working substance?

Ans. The working substance is the medium which receives heat, converts a small portion of it into mechanical energy, and



FIGS. 2,642 and 2,643.—***Adiabatic line.*** When suitable means are not provided for maintaining a constant temperature, or if the action of the gas correspond to that of a piston moving at a high rate of speed, or if the gas act within a cylinder which is a non-conductor of heat, then, during expansion the temperature will fall by an amount equal to the heat units expended in doing external work, and during compression the temperature will rise by an amount equal to the number of heat units of work spent upon it. This mode of expansion or compression is called ***adiabatic*** *since there is no transfer of heat to or from the gas* through the walls of the cylinder, and the external work during expansion is done entirely from the internal energy of the gas. It is obvious, that adiabatic action must be accompanied by a change in the ratio of pressure to volume, therefore, at the end of the piston stroke the pressure will be less than in the case of isothermal expansion by an amount corresponding to the heat expended in doing external work. In the figure, the curve of adiabatic expansion is shown by the line B C. In isothermal action P V =constant, therefore, in order to express the law of adiabatic action, it is necessary that the factor V, be given an exponent greater than unity, and if n, be put to represent this exponent, then $P V^n$ =constant.

rejects the remainder still in the form of heat. The working substance may be a solid, a liquid, or a gas, and the action of the engine may be due to changes of either the form and the volume of the working substance, or to changes of volume alone.



FIGS. 2,644 and 2,645.—***Isopiestic line*** or ***isobar.*** The isothermal and adiabatic lines shown in the preceding diagrams, are not the only thermal lines which may be graphically represented on the P V diagram. For example, if a given quantity of gas be placed in a cylinder and allowed to expand under the influence of heat, or if its volume be increased in any other manner, while its pressure is kept constant, its expansion will be represented by a horizontal line parallel to the line of volumes, and will lie above the latter at a height proportional to the constant pressure maintained above it. Such a line is called an ***isopiestic line*** or ***isobar,*** and corresponds to B C, in the figure. ***Isopiestic expansion*** *occurs during the charging stroke of a gas engine, when the charge is drawn into the cylinder, and* ***isopiestic compression*** *occurs during the exhaust stroke, when the burnt gases are being expelled from the cylinder,* provided that in both cases the area of the valves is sufficient to prevent variations of pressure during the operations which change the volume of the gas.

Ques. What are the most suitable working substances for heat engines?

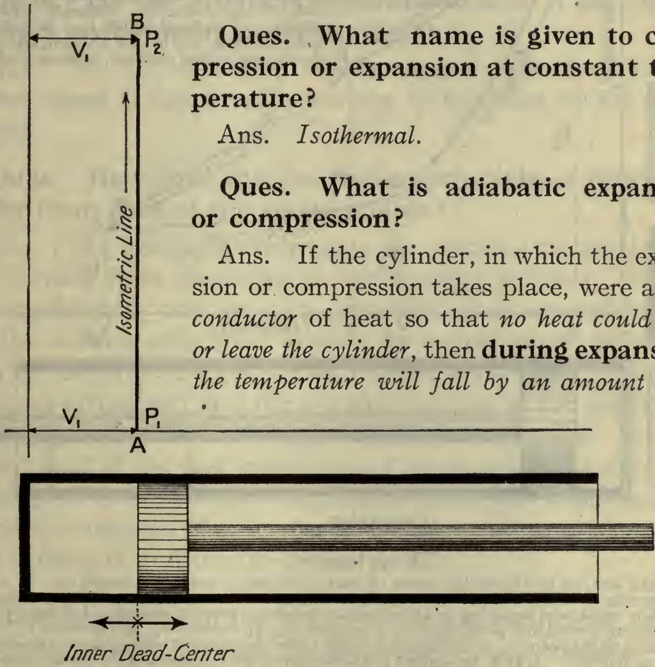
Ans. Gases, because they are almost perfectly elastic, and expand largely under the influence of heat.

Ques. What name is given to compression or expansion at constant temperature?

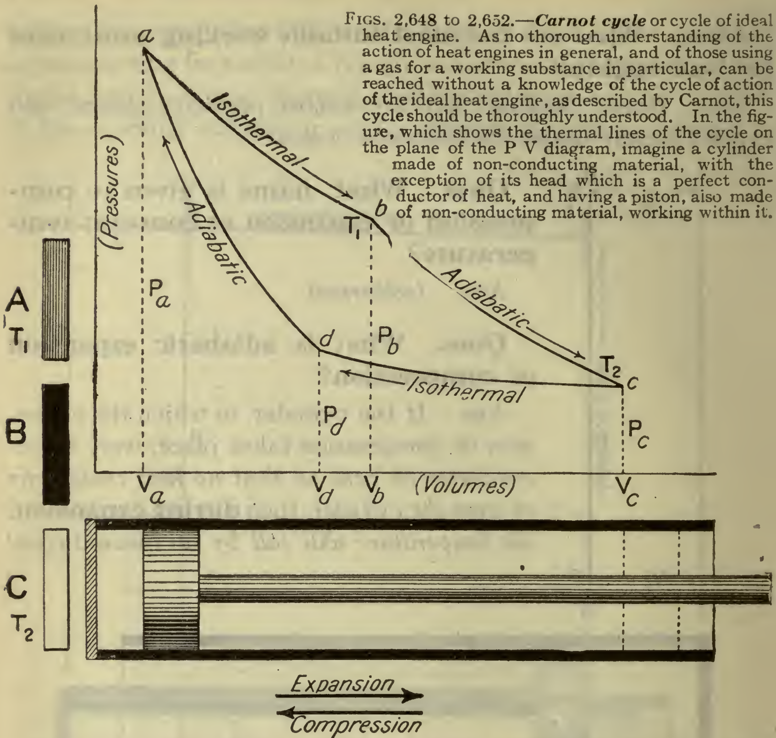
Ans. *Isothermal.*

Ques. What is adiabatic expansion or compression?

Ans. If the cylinder, in which the expansion or compression takes place, were a *non-conductor* of heat so that *no heat could enter or leave the cylinder*, then **during expansion,** *the temperature will fall by an amount equal to the heat units expended in doing external work, and* **during compression,** *the temperature will rise by an amount equal to the number of heat units of work spent upon it.*



FIGS. 2,646 and 2,647.—***Isometric line.*** If a given mass of gas be placed in a cylinder and heated while its volume is kept constant; then, the addition of heat will cause an increase of pressure, and the line traced on the P V diagram will be vertical, and will lie parallel to the line of pressures at a distance proportional to the space occupied by the original volume of the gas as shown by the line A B. Such a line is called an ***isometric line,*** and represents *the condition when the charge in the cylinder of a gas engine is ignited while the piston stands* at its inner *dead center.*



FIGS. 2,648 to 2,652.—***Carnot cycle*** or cycle of ideal heat engine. As no thorough understanding of the action of heat engines in general, and of those using a gas for a working substance in particular, can be reached without a knowledge of the cycle of action of the ideal heat engine, as described by Carnot, this cycle should be thoroughly understood. In the figure, which shows the thermal lines of the cycle on the plane of the P V diagram, imagine a cylinder made of non-conducting material, with the exception of its head which is a perfect conductor of heat, and having a piston, also made of non-conducting material, working within it. Imagine also, an inexhaustible source of heat A, having a constant temperature T_1, a non-conducting cover B, and a cold body C, having a constant temperature T_2, lower than T_1, which is capable of receiving or absorbing an indefinite amount of heat. Let the cylinder contain one pound of a perfect gas, at a temperature T_1, volume V_a, and pressure P_a, corresponding to the point *a*, on the diagram. The cycle of action will be the result of four operations as follows: ***1,*** If A, be applied to the bottom of the cylinder and the piston allowed to move, the gas will expand isothermally at a temperature T_1 taking heat from A, and pushing the piston forward to b, and the pressure will change to P_b. and the volume to V_b. ***2,*** Now, if A, be removed and B, applied to the head of the cylinder, the gas will expand adiabatically, continuing to push the piston forward at the expense of its own internal energy, and its temperature will fall. Allow the gas to expand until its temperature falls to T_2. Then, the pressure will have fallen to P_c and the volume increased to V_c. ***3,*** Now, remove B, and apply C, to the bottom of the cylinder and force the piston backwards. The gas will be compressed isothermally at a temperature T_2, rejecting into C, all the heat due to the work spent upon it by the act of compression. Continue the compression until a point *d* is reached from which a line of adiabatic compression will pass through the point *a*. ***4,*** If C, be now removed and B, again applied to the head of the cylinder, the gas will continue its compression adiabatically, the pressure and temperature rising, and if the point *d* be properly chosen, the cycle will be completed by the restoration of the original temperature T_1, and the original pressure P_a, when the piston reaches the point corresponding to the original volume V_a.

This mode of expansion or compression is called *adiabatic* since there is *no transfer of heat to or from the gas through the walls of the cylinder*, and the external work during expansion is done entirely from the internal energy of the gas.

Ques. Do the products of combustion in a gas engine expand isothermally or adiabatically?

Ans. No; the expansion follows more nearly the adiabatic curve; there is considerable transfer of heat due to the water jacket.

Ques. How does the gas engine principle of operation differ from that of the steam engine?

Ans. The gas engine represents the practical transferance of the furnace from the boiler to the working cylinder. This is accomplished by the use of a permanent gas, atmospheric air, for the working substance, which is intimately mixed with a fuel also in the form of a gas. This mixture constitutes the charge which is introduced into the working cylinder of the engine and there ignited. The resulting combustion liberates the inherent heat of the fuel component of the mixture and applies it directly to every molecule or particle of the air component or working substance, thus increasing its temperature and pressure, and causing it to drive the piston.

Classification.—In the history of the gas engine, the numerous types brought out may be divided into three classes:

1. Those in which the charge is admitted to the working cylinder at atmospheric pressure and ignited without compression;

2. Those in which a charge of the mixture is drawn in by suction of the working piston, and ignited after being compressed by a return stroke of the piston.

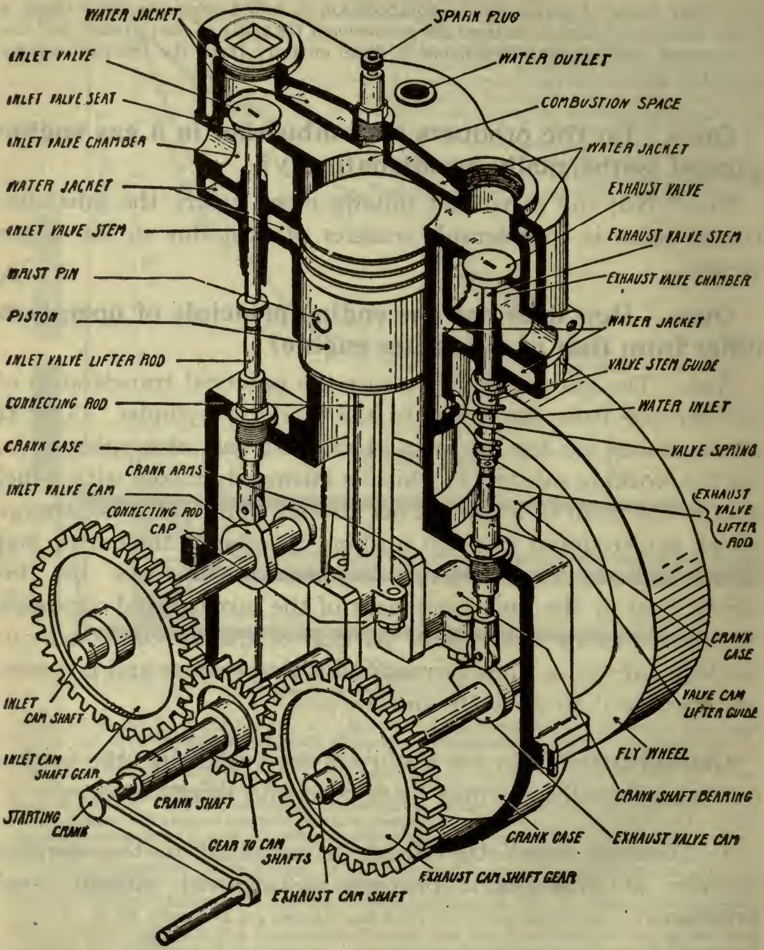


FIG. 2,653.—Sectional view of a four cycle gas engine, showing the valve gear and other working parts. Both inlet and exhaust valves are mechanically operated. The location of the valves diametrically opposite each other requires a separate cam shaft for each. These cam shafts are geared to the engine crank shaft and they make one revolution to every two of the engine. When the inlet valve is operated by a spring and the engine suction, only one cam shaft is necessary, as shown in fig. 2,654.

3. Those in which the *mixture* is first compressed in an auxiliary cylinder, and a *charge* admitted to the working cylinder under a definite constant pressure and ignited when fully admitted.

Cycles of Operation.—All modern gas engines belong to the second class, and may be grouped into two well defined types, according as the cycle of operation is completed in two, or four strokes, the familiar terms

1. Two cycle;
2. Four cycle.

The four cycle engine being more generally used than the two cycle, is the more important and will accordingly be considered first. Fig. 2,653 shows the general arrangement of parts of a four cycle engine.

Ques. What is four cycle operation?

Ans. The term "four cycle" is an abbreviation of *four stroke cycle,* and means that the cycle of the working substance is completed in four strokes of the piston.

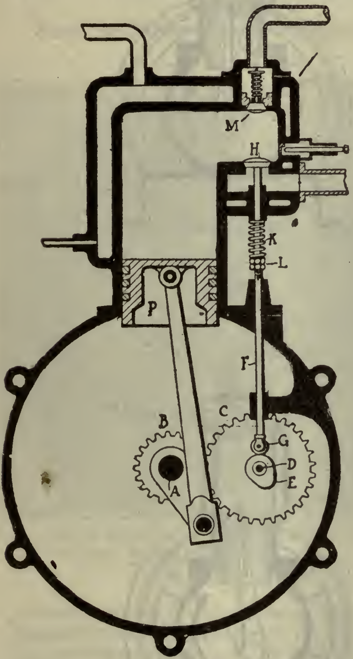
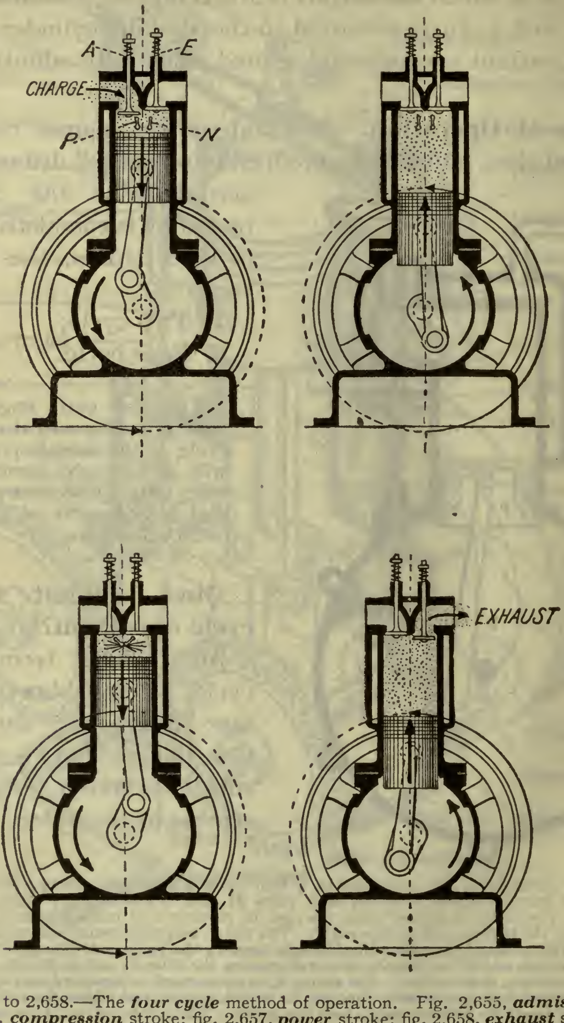


FIG. 2,654.—Illustrating the operation of a four cycle engine with spring inlet valve gear. The figure shows a spring actuated inlet valve M, and a mechanically operated exhaust valve H. The latter is opened when the cam E, revolves and raises against the roller G, which is on the bottom of the lifter rod F. The rod F, extends upward and rests against the bottom of the stem of the valve H, although between the two or at their point of contact are nut and lock nut L, for lengthening or shortening the lifter F, and so to vary the time of opening or closing of the valve. The spring K, is compressed or squeezed together when the valve is opened and immediately the cam E, travels around and allows the roller G, to fall; this spring exerts its pressure and closes the valve. The intake valve M, is automatically opened by the suction of the engine.



FIGS. 2,655 to 2,658.—The *four cycle* method of operation. Fig. 2,655, ***admission*** stroke; fig. 2,656, ***compression*** stroke; fig. 2,657, ***power*** stroke; fig. 2,658, ***exhaust*** stroke. A, is the admission valve; E, exhaust valve; N, negative electrode; P, positive electrode of igniter.

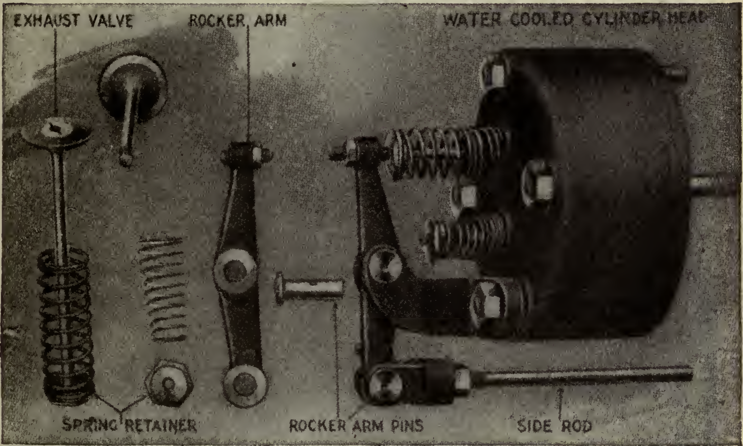
Ques. What name is given to the four strokes of the cycle?

Ans. Admission, compression, expansion, and exhaust.

Ques. What other names given to the admission and expansion strokes?

Ans. The terms *suction* and *power* are sometimes used.

Since the four cycle engine represents the most generally used gas engine, a consideration of its indicator diagram will aid in ascertaining the general characteristics of gas engine cycles. In the four cycle engine, the cycle of the working substance, as previously mentioned, is completed in four strokes according to the following sequence: admission takes place during the first forward or suction stroke, compression during the following return stroke, expansion during the second forward or power stroke, and exhaust during the second return or exhaust stroke; in tabular form:



FIGS. 2,659 to 2,665.—Valve detail of Gray four cycle engine and view of cylinder head showing valve gear (spring inlet valve).

1. First stroke (outward) *admission;*
2. Second stroke (inward) *compression;*

3. Third stroke (outward) *expansion;*
4. Fourth stroke (inward) *exhaust.*

Fig. 2,666 shows a theoretical diagram of a four cycle engine;

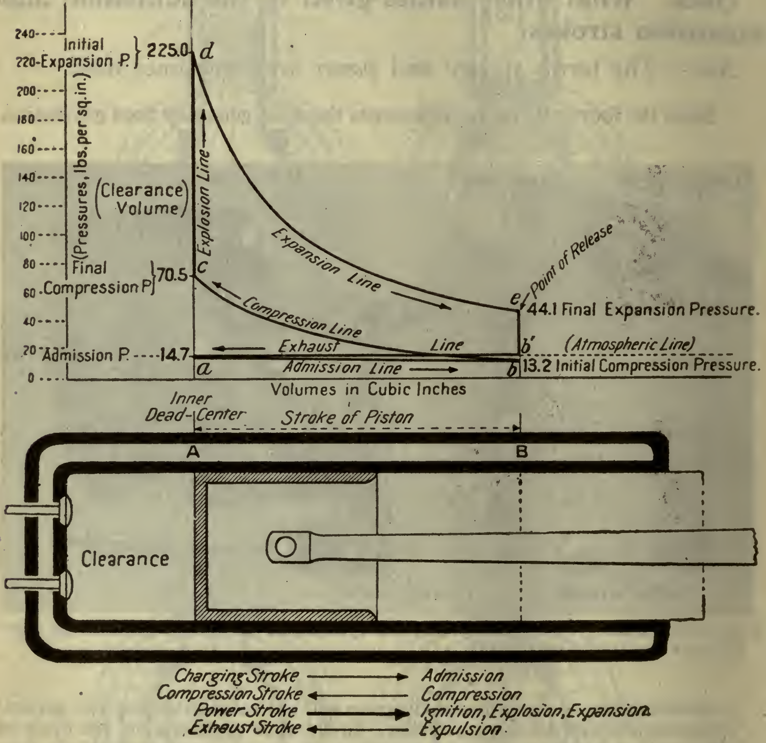

240
Initial Expansion P. 225.0 d
220
200
180
160
140
120
100
80
60
40
20
0
(Pressures, lbs. per sq. in.)
(Clearance Volume)
Explosion Line
Final Compression P. 70.5 c
Expansion Line
Point of Release
e
44.1 Final Expansion Pressure.
Compression Line
Admission P. 14.7
Exhaust Line
b'
(Atmospheric Line)
a
Admission Line
b
13.2 Initial Compression Pressure.
Volumes in Cubic Inches
Inner Dead-Center
Stroke of Piston
A
B
Clearance
Charging Stroke → Admission
Compression Stroke ← Compression
Power Stroke → Ignition, Explosion, Expansion
Exhaust Stroke ← Expulsion

FIG. 2,666 and 2,667.—***Theoretical indicator diagram*** of a four cycle gas engine. The assumed values for temperature, volume, and pressure are a fair average of those occurring in the cylinder under actual working conditions.

the assumed values for temperature, volume, and pressure, however, do not correspond to the maxima and minima of such as may be derived from theoretical computations, but represent

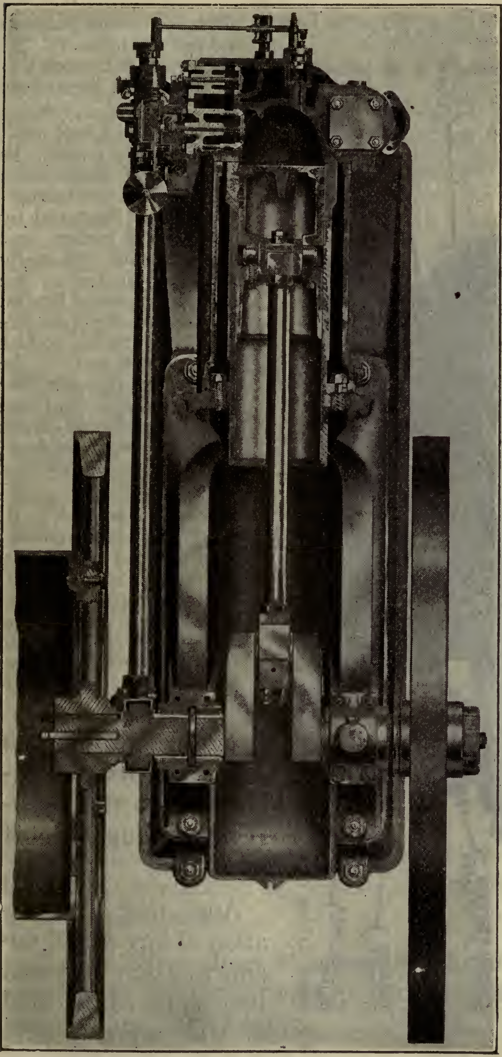


FIG. 2,668.—Sectional view of Otto gas engine showing cylinder, combustion chamber, valve gear, etc.

values which are a fair average of those occurring in the cylinder of a gas engine, operating under actual conditions.

In the diagram, the cycle begins at A, with the piston at *a*. During admission, the piston moves to B, the conditions within the cylinder during the movement being *ab*. This line is drawn below the line of atmospheric pressure, because, due to suction, the pressure in the cylinder is less than that of the atmosphere. At B, the inlet valve closes, thus completing admission.

The piston now returned from B, to A, compressing the previously admitted charge into the clearance or combustion chamber; the curved line *bc*, shows the gradual increase in pressure, due to compression, from about a pound below atmospheric pressure to 70½ pounds at the end of the stroke. At this instant ignition of the compressed charge takes place, and the pressure suddenly jumps to 225 pounds, corresponding to the initial expansion.

The piston again moves outward from A, to B, making the power stroke; the products of combustion expand with the increasing volume displaced by the moving piston, and the pressure gradually falls, as indicated by the curved expansion line *de*. As shown, the pressure falls from 225 lbs. to 44.1 lbs. at the end of the stroke. At this point the exhaust opens and the pressure suddenly drops to that of the atmosphere, shown by the line *eb'*.

During the final stroke of the cycle from A to B, the exhaust valve remains open and the burnt gases are expelled from the cylinder at approximately atmospheric pressure, as represented by the line *ba*, thus completing the cycle.

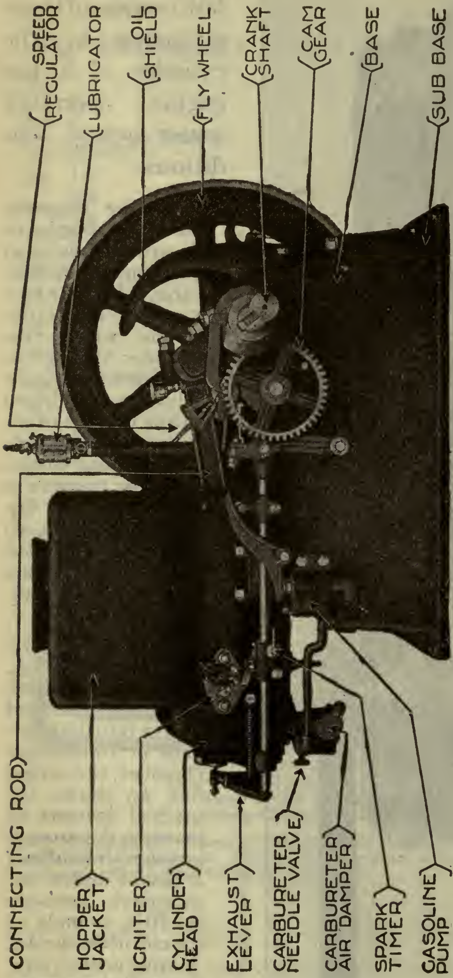


FIG. 2,669.—Alamo four cycle engine with fly wheel removed, showing parts.

Ques. Could an engine operate according to the theoretical diagram of fig. 2,666?

Ans. *No*; the operation of any actual engine *can only approach* the conditions indicated in the diagram, since it is limited by mechanical and physical restrictions.

Ques. Of what use, then, is a theoretical diagram?

Ans. In designing a gas engine, its probable performance can be approximated by first drawing a theoretical diagram corresponding to the given data, and then inscribing an actual working diagram by making proper allowances for the various losses that are known by test to take place in engines of similar type already built.

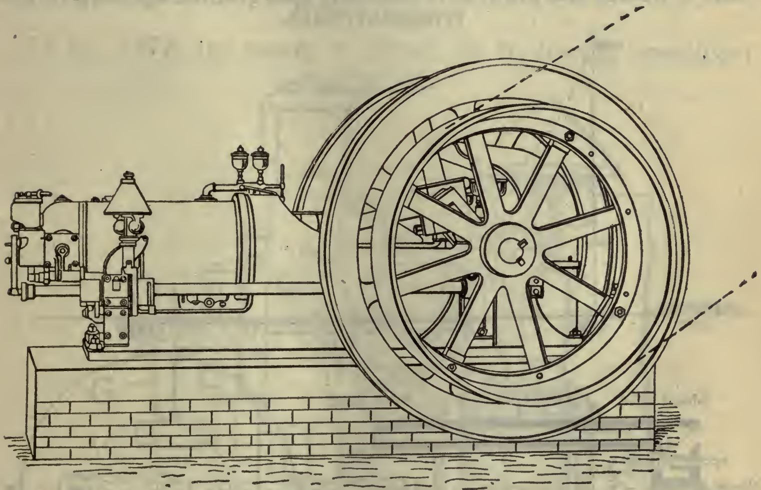
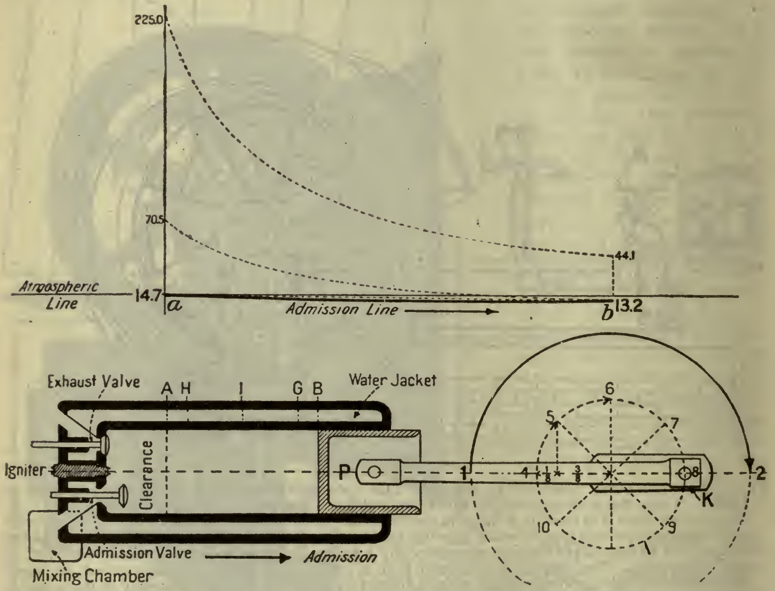


FIG. 2,670.—50 horse power Otto engine. The construction of Otto engines was begun in 1867. It was the first practically successful engine of the four cycle compression type. At the present time it is built in a large range of sizes. The frame is of substantial pattern with well rounded corners and a broad bearing on the foundations. The cylinder overhangs the frame, and in the larger sizes is supported by a heavy cylinder foot immediately in front of the valve gear. The engine is equipped with a long connecting rod which serves to lessen the vertical thrust upon the piston, thus adding to the tightness and durability of both piston and cylinder by diminishing friction and consequent wear.

Ques. Why does the operation of an actual engine vary from the ideal performance represented by the theoretical card, fig. 2,666?

Ans. In nature, nothing can take place *instantaneously*, that is, an *interval of time* is necessary for a physical change to take place, hence there can be no vertical lines as *cd*, or *eb'*, on a working diagram. Again, time is required for the opening of the exhaust valve, involving that it *begin to open* before the end of the stroke, that is before the point *e*, is reached; this gradual opening of the valve is accompanied by a gradual instead of an abrupt reduction of pressure. Numerous other causes tend to produce variations between the actual and ideal diagrams.



FIGS. 2,671 and 2,672.—***Admission stroke.*** The admission line at *ab*, sags below the theoretical on account of frictional resistance encountered by the incoming charge.

The observation of these variations, the determination of their causes, together with an assignment of their probable values, constitute a working theory of each particular machine, upon which designers may rely. Such a theory, founded upon observation and experiment, is only to be deduced from long continued tests carried out by many persons, with painstaking care, and under all conceivable conditions of operation. In order to point out the numerous variations and their causes, it will be well to compare the actual diagram with the theoretical, by drawing the former stroke by stroke. In this process, the theoretical diagram of fig. 2,666 is shown in each figure in dotted lines for direct comparison and the actual diagram which is drawn in full lines.

Admission

In fig. 2,672, the piston is shown at B, having completed the admission stroke; at the beginning of admission the pressure in the cylinder is that of the atmosphere, 14.7, and

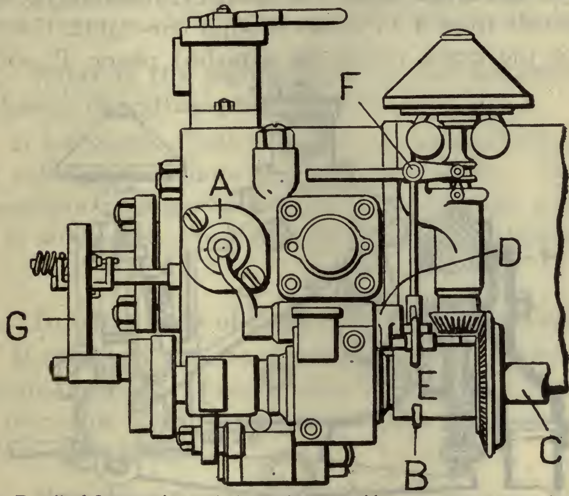


FIG. 2,673.—Detail of Otto engine cylinder, valve gear side. ***In operation,*** when gasoline is used, it is delivered by the gasoline pump driven by an eccentric on the side shaft, into the stand pipe from which it gravitates to the gasoline valve A, which regulates the quantity of gasoline and the time of its admission, so as to supply automatically the proper mixture of air and gas according to the requirements of the varying load. The valve A, is operated by the cam B, on the side shaft C, the cam actuating a rocker shaft D, the lower arm of which is fitted with a long pin which carries the gasoline roller E. This roller has a free lateral movement which is governed by the vertical arm of the bell crank F, connected with the governor, a very slight movement of which is sufficient to place the roller in or out of engagement with the cam on the shaft. The speed of the engine is regulated by the governor which changes the position of the gasoline roller, so that the fuel is admitted only when the speed is normal or slightly below it. The governor is of the fly ball weighted type. Ignition is effected by an electric spark, the igniter being operated by the igniter lever G, actuated by the side shaft. The electrodes of the igniter are tipped with platinum, and the current is derived from a primary battery.

is assumed to drop to 13.2 lbs. at the end of the stroke. This is represented in the theoretical diagram, fig. 2,671, by the dotted *straight* line *ab*, and in the actual diagram by the *curved* full line *ab*.

Ques. Why does the full admission line *ab* of the actual diagram, fig. 2,671 curve or sag below the dotted straight line ab?

Ans. In fig. 2,672 while the crank K, revolving at uniform speed, travels from 4 to 5, or through one-eighth the circumference of the crank circle, the attached piston P, will move forward from A to H, or through approximately one-eighth of its stroke. Again, while the crank passes from 5 to 6, a distance also equal to one-eighth of the circumference of the crank circle, the piston will move from H to I, about three-eighths of its stroke.

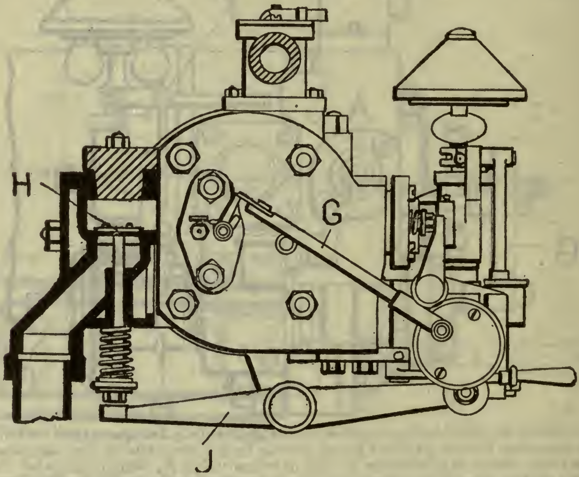


FIG. 2,674.—Detail of Otto engine cylinder; sectional end view. ***As shown,*** the exhaust valve H, is operated by the exhaust lever J, fulcrumed beneath the cylinder, and actuated by a cam on the side shaft. The cylinder, cylinder head, and exhaust valve box are amply water jacketed. The piston and the cross head pin are lubricated by means of special oil cups placed on the cylinder near the front end. The crank pin is oiled by means of a wiper fed by a sight feed cup, and the main bearings by ring oilers which dip into an oil reservoir beneath the bearing.

In other words, the uniform rotary motion of the crank causes great variations in the speed of the piston at different points of its stroke, the speed of the piston being more rapid while the crank passes from 5 to 7, and from 9 to 10, than while it passes from 10 to 5, and 7 to 9. The movement of the incoming charge as it flows from the carburetor through the manifold and valve opening is opposed by frictional resistance which increases with its speed. Hence, the increasing speed of the piston during the first part of the stroke causes an increase of speed of the charge, this in turn involves a drop of pressure within the cylinder corresponding to the greater suction required to overcome the excess frictional resistance. Thus, it is seen that the admission line falls below the dotted line during the first part of the stroke, and since in the latter part of the stroke the reverse conditions obtain, the admission line gradually approaches the dotted line.

Ques. What is the significance of the admission line falling below the atmospheric line?

Ans. It represents a loss of power, as measured by the area enclosed between these lines and the stroke limits. That is, if the atmospheric pressure be in excess of that in the cylinder, it will act in a direction directly opposed to the movement of the piston.

Ques. In what class of engine is this loss a maximum?

Ans. It is greater with an automatic inlet valve than with a mechanically operated valve, because the suction must be sufficient to overcome both the resistance of the valve and that due to the flow of the charge.

Ques. How does this loss vary?

Ans. It increases with the speed of the engine.

Ques. In what other way does this affect the engine operation?

Ans. It determines the quantity of the mixture admitted, and has a direct bearing on the compression.

Ques. What point, in regard to the admission line, is sought in designing an engine?

Ans. It is desirable, at normal speeds, that it lie as close as possible to the atmospheric line, consistent with reasonable proportions and arrangement of the mechanical details.

Ques. What features of design tend to accomplish this?

Ans. Short and direct manifold passages without abrupt or tortuous bends; proper location of valves and adequate cross sectional area of manifold and port opening, so that the proper rate of flow is obtained at normal speeds.

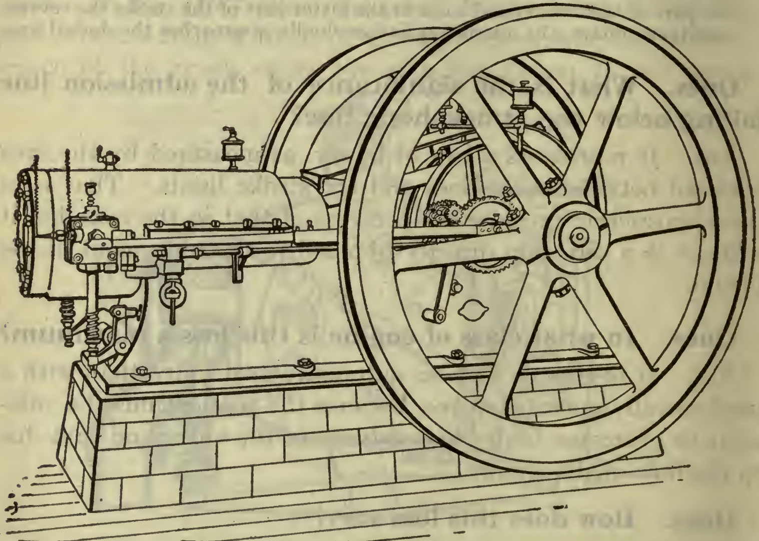


FIG. 2,675.—Foos gas engine, valve gear side. All the working parts are assembled on one side of the cylinder. The design is the same for all sizes, the cylinders being bolted to the beds by long ways which project from the sides.

Ques. What is the proper speed through the manifold and admission valve?

Ans. Not over 4,000 or 5,000 feet per minute.

Example—What area of valve opening is required for a 5×5 engine designed for a normal speed of 600 revolutions per minute, and a rate of flow of the mixture through the valve of 4,000 feet per minute?

The proper valve is found from the following formula:

$$\textit{valve area} = \frac{\textit{piston area} \times \textit{piston speed}}{4{,}000} \quad \ldots\ldots\ldots\ldots \quad (1)$$

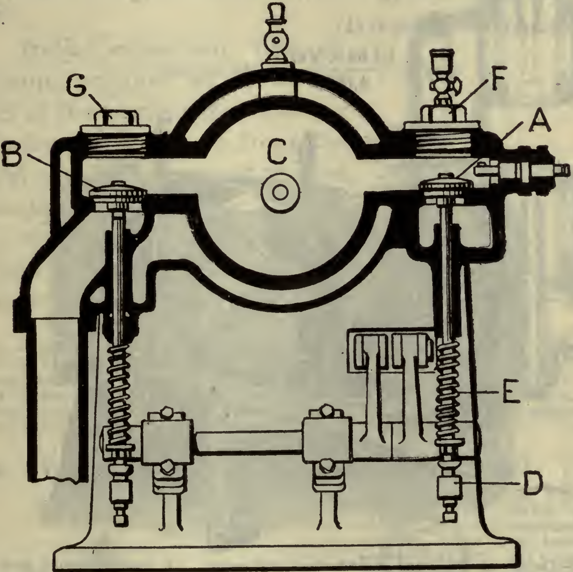


FIG. 2,676.—Foos gas engine, section view through cylinder and valve chambers. In the figure it will be noted, that the admission valve A, and the exhaust valve B, are placed in separate water jacketed castings on opposite sides of the cylinder, and communicate by means of large ports with the combustion chamber C. These valves are of the vertical poppet type, and are positive in their action. As the weight of the valve acts in the direction of its motions it returns naturally to its seat and maintains a tight contact even without the aid of a spring. Suction in the cylinder is not relied upon to open the admission valve; it being liften positively by the lever D, and closed by its own weight aided by the action of the strong spring E. Although the valves are in separate removable boxes it is not necessary to detach these boxes in order to remove the valves. The admission valve can be taken out by unscrewing the plug F, and the exhaust valve by removal of the plug G. The valves can be ground in when necessary by the use of an ordinary brace and bit. The valve gear is operated through steel cams driven by machine cut gears; and by adjusting, with a common wrench, the screws carried by the lift lever D, any wear or looseness in the valves may be taken up. The governor is of the centrifugal type. It regulates the supply of fuel by cutting out the charges beyond those required to maintain uniform speed. The igniter is of the electric wipe contact type, and is located in the admission chamber directly over the admission valve.

In the formula, the piston area is taken in square inches, and the speed in feet.

The area of a 5 inch piston is

$$.7854\ D^2 = .7854 \times 5^2 = 19.64 \text{ sq. ins.}$$

The piston speed is equal to

$$\frac{2 \times \text{stroke} \times \text{revolutions per minute}}{12} = \frac{2 \times 5 \times 600}{12} = 500 \text{ ft.}$$

Substituting these values in (1),

$$\text{area} = \frac{19.64 \times 500}{4,000} = 2.46 \text{ sq. ins.}$$

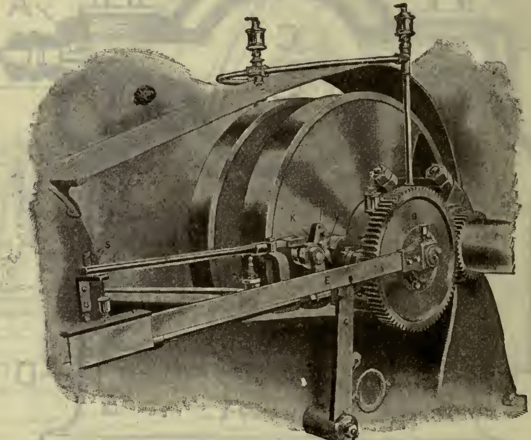


FIG. 2,677.—Detail of Foos gas engine cams and gears. A large reducing gear G, turned by a pinion on the crank shaft operates the igniter rod E. The two cams U and V, on the opposite end of the gear shaft operate the inlet and exhaust valve. Another cam secured to the larger gear G, controls the fuel valve through the lever C. ***The auxiliary cam*** in the position indicated by K, is furnished on larger engines to aid in starting. It holds the exhaust valve open during enough of the compression stroke to relieve the pressure in the cylinder. This cam is controlled by the hand lever F, held in position by the pin S.

Having calculated the area, the diameter, lift, etc., of the valve are determined as explained under carburetters.

Ques. What bearing has admission on the power developed by the engine?

Ans. The amount of power developed depends upon the amount of fuel mixture admitted; the greater the charge, the greater the power.

Ques. What condition tends to diminish the amount of the charge?

Ans. Initial expansion, that is, the incoming mixture as it strikes the walls of the combustion chamber and cylinder, absorbs heat from the latter and expands; hence, at the end of the suction stroke, there is a less amount of mixture in the cylinder than would have entered had the temperature remained constant.

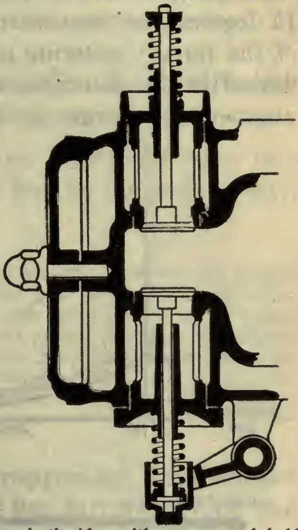


Fig. 2,678.—Olds valves. As shown, the valves are vertical. Both valves are in cages and can be removed from the engine without changing the timing of valves or spark. The joint between the valve cage and the cylinder is made by a small gasket faced on both sides with copper and held in a recess in such a way as to prevent the gasket blowing out.

Ques. When does the admission valve close?

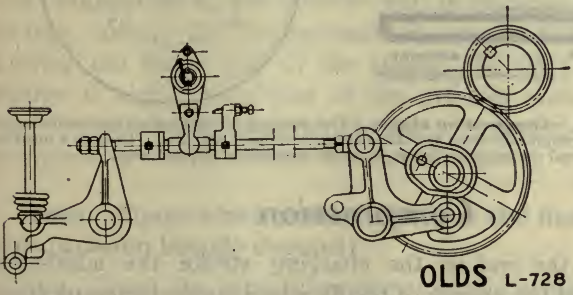
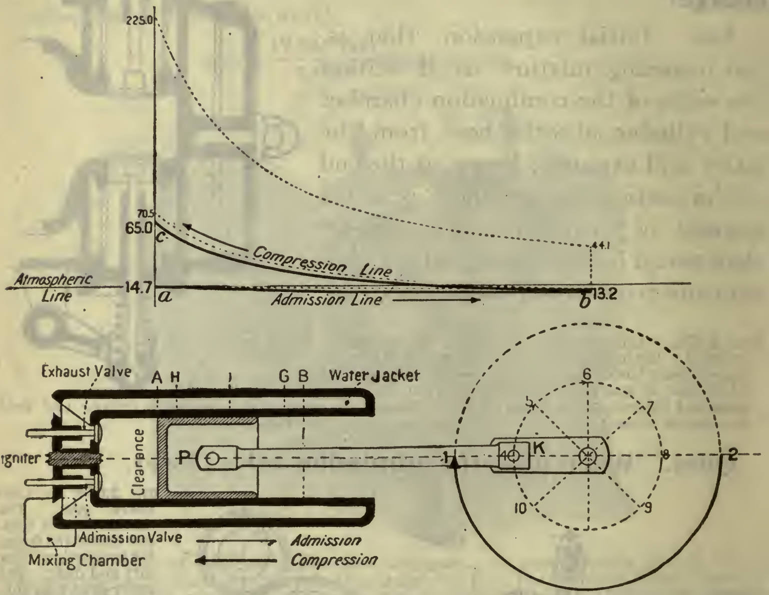


Fig. 2,679. — Olds valve gear. The mechanism is located inside the frame for protection from dirt and dust. The lower gear dips into the oil bath in the bottom of the frame. The cam is keyed to the larger gear and both cam and gear are keyed together on a hardened steel pin. The cam roller is lubricated by dipping into the oil bath in the bottom of the frame.

Ans. In general, it should *begin* to close just before the end of stroke, so that it will be shut when the piston reaches the end of the stroke. With the engines designed to run at high speed, the closing of the valve is sometimes delayed as much as 10 or 15 degrees past the center, the theory being that the momentum of the rapidly entering mixture will cause a continuance of its flow after the piston has reached the end of the stroke, and thus augment the charge.



FIGS. 2,680 and 2,681.—*Compression stroke,* The working conditions within the cylinder are such that the compression does not take place according to any law. There is a marked variation of the actual compression line *bc*, from the theoretical (dotted line).

Compression

At, or near the end of the charging stroke the admission valve closes and the energy of the fly wheel moves the crank from position 2 to 1 (fig. 2,681), which, in turn, pushes the piston from B, to A, thus compressing the charge into the clearance space.

This results in raising both the pressure and the temperature of the charge. The gradual increase of pressure in the cylinder due to compression is indicated by the full compression line.

Ques. Why does the full compression line fall below the dotted line of the theoretical diagram?

Ans. Because the compression is not adiabatic; there is a loss of pressure due to the absorption of heat by the jacket water, and to more or less leakage.

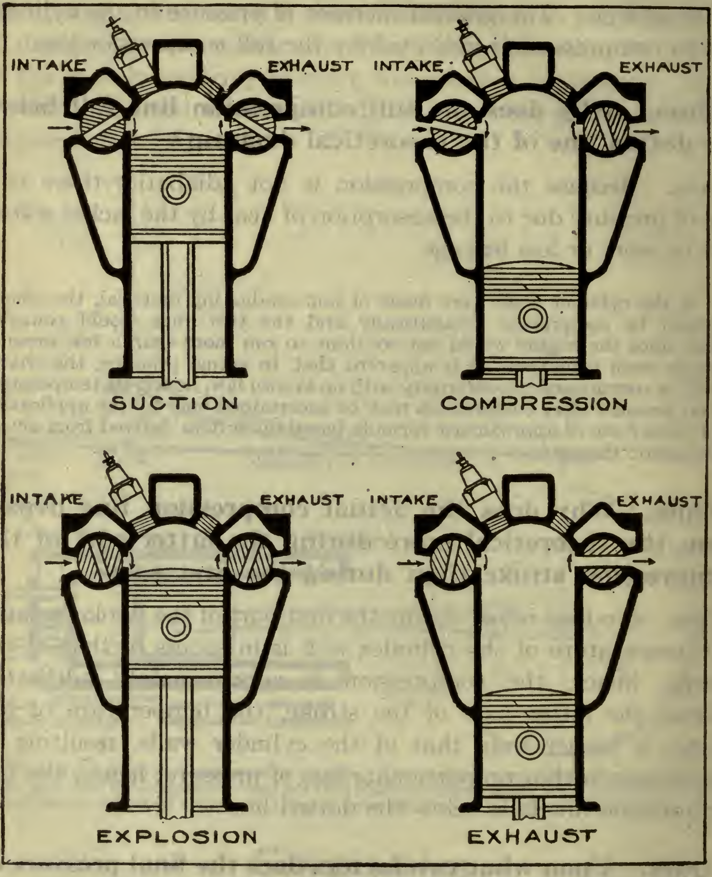
If the cylinder walls were made of non-conducting material, the charge would be compressed adiabatically and the two lines would coincide, but since the engine would not continue to run more than a few minutes under such conditions, it is apparent that, in actual practice, the charge will be compressed in conformity with no known law, hence, its temperature and pressure after compression may be ascertained only by the application of some form of approximate formula based upon data derived from actual indicator diagram.

Ques. Why does the actual compression line depart from the theoretical more during the latter part of the compression stroke than during the first part?

Ans. No heat is lost during the first part of the stroke because the temperature of the cylinder wall is in excess of that of the charge, hence, the compression is approximately adiabatic. During the latter part of the stroke, the temperature of the charge is higher than that of the cylinder walls, resulting in loss of heat with a proportionate loss of pressure; hence, the full compression line falls below the dotted line.

Ques. Upon what two factors does the final pressure of compression largely depend?

Ans. Upon the initial pressure (*i.e.*, the pressure at the beginning of the compression stroke), and the percentage of clearance, assuming leakage.



FIGS. 2,682 to 2,685.—Diagram illustrating valve action of Speedwell (automobile) rotary valve engine. The valves, as shown, consist of cylindrical shafts in the head of the engine. These shafts are slotted and in rotating, register with ports in the cylinder walls thus performing their functions of intake and exhaust. The rotary valve movement is a continuous revolution in one direction. The diagrams illustrate different positions of rotary valves at the beginning of each of the four cycles. The arrows within the cylinders indicate the direction of rotation of rotary valves; the arrows outside indicate the direction of live gas passing in and the exhaust gas passing out.

Example.—If an engine have 33 per cent clearance space, and the charge be admitted at 13 lbs. absolute pressure, and 90° Fahr., what is the final pressure and temperature of compression under the usual working conditions?

The accompanying table has been prepared for determining the compression pressure and temperatures which usually occur in gas engines.

Table of Pressure and Temperature and Coefficients

(Compression)

C	P	T	C	P	T
3.	4.407	146.89	4.	6.498	162.45
3.05	4.506	147.74	4.1	6.718	163.86
3.1	4.606	148.58	4.2	6.940	165.25
3.15	4.707	149.42	4.3	7.164	166.62
3.2	4.808	150.25	4.4	7.390	167.96
3.25	4.910	151.06	4.5	7.618	169.29
3.3	5.011	151.87	4.6	7.847	170.59
3.35	5.115	152.67	4.7	8.078	171.88
3.4	5.217	153.47	4.8	8.311	173.17
3.45	5.322	154.25	4.9	8.546	174.40
3.5	5.462	155.03	5.	8.783	175.64
3.55	5.531	155.80	5.1	9.020	176.87
3.6	5.637	156.57	5.2	9.260	178.07
3.65	5.742	157.32	5.3	9.501	179.26
3.7	5.848	158.08	5.4	9.744	180.44
3.75	5.956	158.82	5.5	9.988	181.60
3.8	6.064	159.56	5.6	10.234	182.75
3.85	6.171	160.29	5.8	10.730	185.01
3.9	6.280	161.02	6.	11.233	187.22

These pressures and temperatures are obtained by means of the table as follows:

Final compression pressure = initial pressure × P . . . . (1)

Final compression temperature = intial temperature abs. $\times \frac{T}{100}$ (2)

In these two formulæ:

P = pressure factor for the given ratio of compression C;

T = temperature factor for the given ratio of compression C;

For a clearance space of 33⅓ per cent of the piston displacement the compression ratio, C, is:

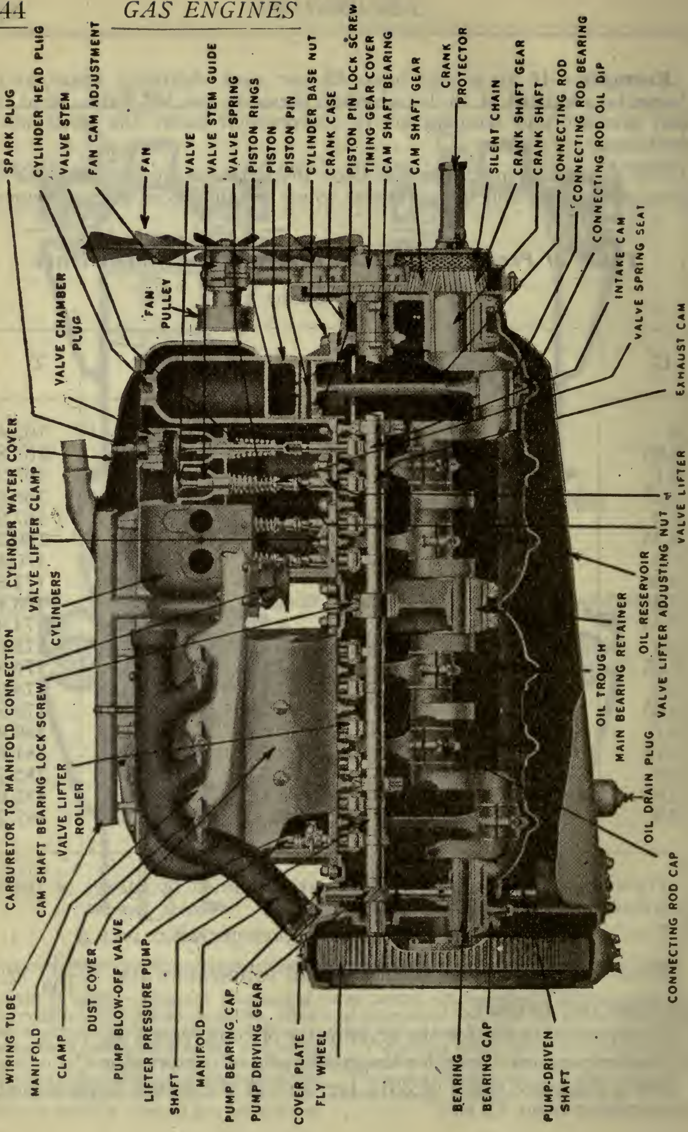


FIG. 2,686.—Sectional view of Lozier six cylinder automobile engine.

$$\frac{1+33\frac{1}{3}}{33\frac{1}{3}}=4;$$

the pressure factor corresponding to C=4, from table is given as 6.498. Substituting these values in (1)

Final compression pressure $=13\times6.498=84\frac{1}{2}$ lbs.

Similarly, substituting in (2),

$$\text{Final compression temperature}=(90+459.2)\times\frac{162.45}{100}=892^\circ$$

Ignition.—The timing of the spark should be varied to suit the running conditions. In general, ignition should occur a little before the end of the compression stroke, in order that the explosion will take place at practically constant volume, and attain the maximum pressure at the beginning of the stroke; early ignition is essential in some instances to secure complete combustion before the exhaust valve opens.

The term combustion means *rapid oxydation*, that is, combustion is a union of the elements which constitute the fuel, with oxygen, the phenomenon being accompanied by an evolution of heat.

Ques. What are the elements of a fuel?

Ans. Hydrogen and carbon.

Ques. Under what conditions of operation is the maximum temperature reached that is possible with a given mixture and compression?

Ans. When the spark is advanced sufficiently that the maximum temperature of combustion occurs when the piston is at the beginning of the stroke or dead center.

Ques. How is the necessary oxygen for combustion supplied?

Ans. By mixing the fuel with air which contains oxygen.

Ques. How much air is required?

Ans. The most efficient proportion of air to give perfect combustion depends on the kind of fuel and conditions under which combustion takes place. For automobile engines, a variable mixture of gasoline vapor and air is used, in proportions ranging between 1 to 6 and 1 to 11, depending on the immediate conditions of engine operation.

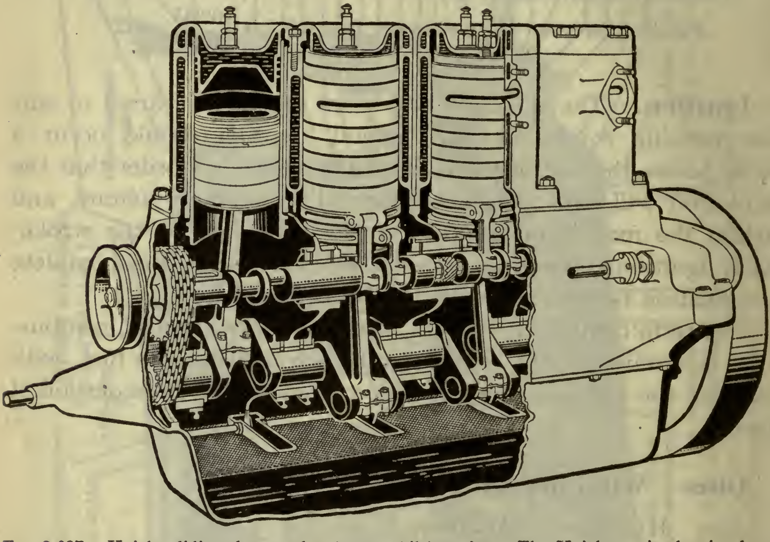


FIG. 2,687.—Knight sliding sleeve valve (automobile) engine. The Knight engine has in place of poppet valves *two sliding telescopic sleeves*; these contain the ports and perform the valve duties, being driven by short connecting rods from the half speed shaft at the side. ***In operation,*** the right hand ports of the sleeves register with each other on the suction stroke, exposing a large opening through which the charge of fuel mixture enters. At the beginning of the compression stroke, the sleeves slide, one up and one down, thus closing the ports for compression. All ports remain closed until the end of the power stroke, when the left hand ports open and register with each other, leaving a large exit for the exhaust of the burnt gases. The advantages claimed for this valve gear are large port openings, and noiseless operation; the above cut illustrates the Columbia design of the Knight engine. The moving parts are lubricated in the usual way, the only precaution taken being to insure an even distribution of oil by cutting a spiral groove and boring a number of holes in each sleeve. A feature of the lubrication is the automatic oil pans below the cranks which by means of a connection with the throttle lever are raised as the throttle is opened, thus increasing the amount of lubrication as the load or speed is increased.

Ques. What effect has the proportion of air on the explosion pressure?

Ans. The explosion pressure decreases in ratio with an increase in the proportion of air.

Ques. What conditions determine the explosion temperature of the charge?

Ans. The maximum temperature depends on: 1, the calorific value of the fuel, 2, the quality of the mixture; 3, the time of ignition, 4, the piston speed, 5, degree of cooling, and 6, amount of leakage.

Ques. How is the explosion temperature affected by the rate of combustion?

Ans. The more rapid the combustion, the higher the explosion temperature.

The chemical reaction due to complete combustion may take place gradually, as in slow combustion, or very rapidly, as in the case of an explosion. Since the calorific value of a fuel depends on its contained heat, and is not affected by the rate of combustion, high temperatures may be attained with fuel of low calorific value by causing the combustion to take place rapidly.

Ques. What are the laws which govern the rate of combustion?

Ans. These laws, as determined experimentally, are as follows: 1, in any mixture the rate of combustion is constant for a given temperature before ignition. 2. The rate of combustion increases with the increase of temperature of the mixture before ignition. 3. The rate of combustion varies with the proportion of the air and fuel components of the mixture.

It is evident that, according to the first two laws, the rate of combustion of an explosive mixture may be made to vary by artificial means; for example, by *compression*, thus making the temperature of combustion partly dependent upon the cycle of the machine. This fact clearly demonstrates the value of compression in its relation to the thermal efficiency of the engine.

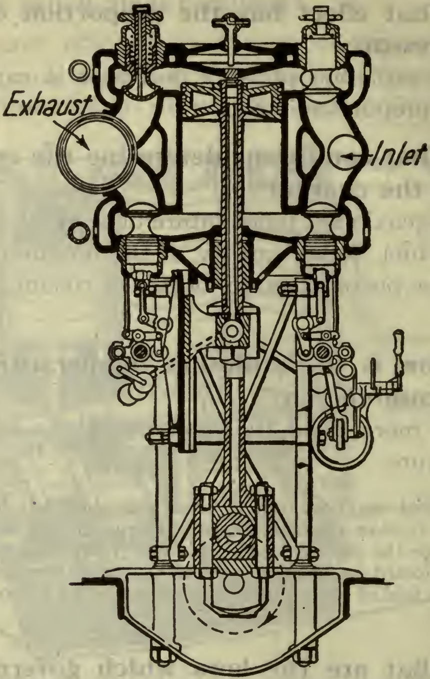
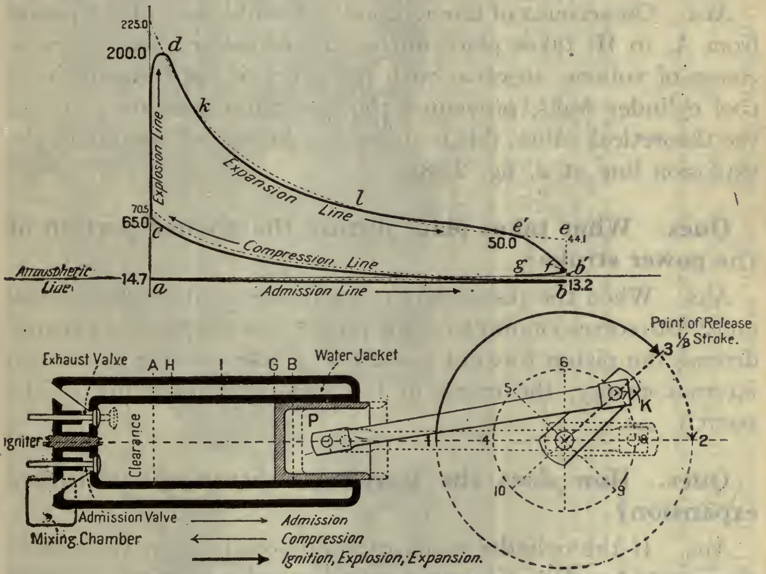


FIG. 2,688.—Standard double acting (four cycle) marine engine. All the valves are positively operated. In addition to the usual water jacketing of the cylinder, the pistons are cooled by circulating water through the cross head, up a tube in the piston rod, and down a return circuit around the tube. ***The engine is started*** by compressed air at a pressure of 250 pounds per sq. in., primarily pumped up into two or more air tanks running along the side of the engine room, by means of small auxiliary gasoline engines which not only drive the air compressor, but also the bilge pumps, and a small dynamo for lighting purposes and for charging the storage batteries. The engine is arranged so as to form two units of three cylinders each, and the three after cylinders only, are connected to the compressed air tanks; when the reverse lever is thrown to the first notch off the center, either forward or backward, the cam shaft is shifted to such a position, that the cams operating the compressed air valves on these three cylinders will lift their exhaust valves once every revolution instead of every other revolution. The three forward cylinders always operate on gasoline, the shifting of the cam shaft having the effect of governing the direction of rotation of the engine only. With the lever in this position, the compressed air is admitted to the three after cylinders and the engine allowed to make two or three revolutions. The lever is then pushed on to the next notch, cutting out the compressed air valves and adjusting the cam to trip the valves in the regular four cycle sequence. Means are also provided by which all the lower exhaust valves can be locked open, and all the lower inlet valves can be locked closed, by a single motion of the lever, thus converting the engine into a single acting engine developing half power. Furthermore, the two units can be unclutched, so that the three after cylinders can be operated alone as a three cylinder single acting unit, thus reducing the power developed to one-quarter the full power of the engine.

According to the third law, the mixture may be varied to suit the immediate conditions of engine operation. Thus, when the engine is heavily loaded, a rich mixture may be used, which, on account of the prolonged combustion, augments the mean pressure during expansion. Again, when the engine is running at high speed but with a light load, a lean mixture, which burns faster, is used to obtain complete combustion at release.

Ques. What is the effect of too little or too much air?

Ans. The mixture will either explode imperfectly or fail to explode.



FIGS. 2,689 and 2,690.—***Power stroke.*** The expansion line varies from the theoretical in a similar manner to that of the steam engine. Pre-release begins at e′, and continues to *b′*.

Expansion

Figs. 2,689 and 2,690 show the action of the working substance during the power stroke. In the diagram, the position of the explosion line indicates that the spark in this instance was retarded to the end of the compression stroke and that the combustion was almost instantaneous, the maximum pressure attained being 200 lbs., corresponding to an initial expansion temperature of 2,195° Fahr. It should be noted that the theoretical pressure is 225 lbs., as shown by the dotted line.

Ques. Why is the actual pressure of combustion obtained less than the theoretical?

Ans. On account of late ignition. A movement of the piston from A, to H, takes place during the explosion, hence, this increase of volume, together with the effect of the comparatively cool cylinder walls, prevented the maximum pressure reaching the theoretical value; this is shown by the curved portion of the explosion line at *d*, fig. 2,689.

Ques. What takes place during the greater portion of the power stroke?

Ans. When the piston has reached the point H, (figs. 2,689 and 2,690) corresponding to *d*, the products of combustion expand, driving the piston forward toward B, at the expense of its own internal energy, the crank in the meantime revolving to the point 3.

Ques. How does the working substance act during expansion?

Ans. If the cylinder walls were non-conducting, the expansion would be adiabatic, but in actual practice a variety of conditions causes the expansion line not only to deviate from the path of an adiabatic line, but also from that of any calculated one.

Ques. Why does the actual expansion line fall below the theoretical, between points *k* and *l*, and above, between points *l* and *e′*, fig. 2,689?

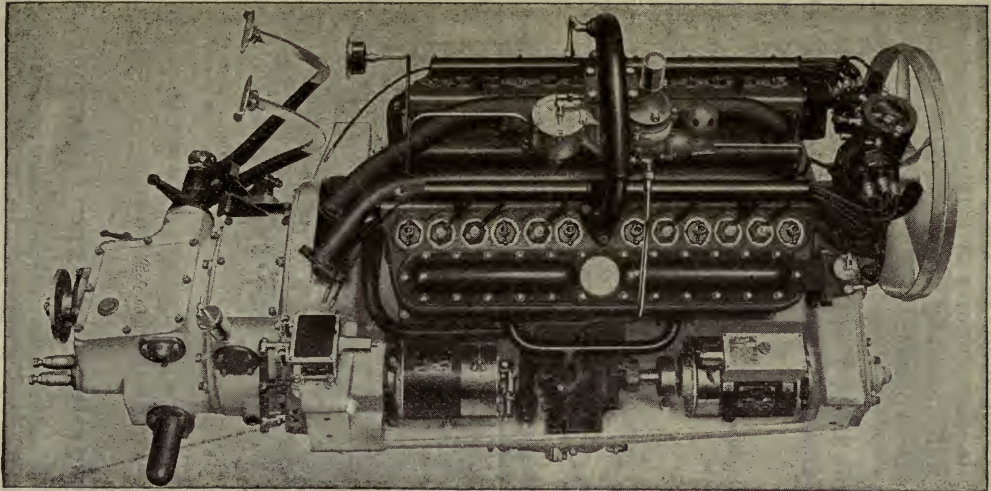


FIG. 2,691.—Packard twelve cylinder (twin six) automobile engine, illustrating multi-cylinder development. The cylinders are arranged six on a side at an included angle of sixty degrees. Displacement 35.5 cu. ins. per cyl. Speed up to 3,000 R.P.M. Bore, three inches; stroke, five inches. Cylinders are L-head type and cast in two blocks of six, with gas intake header, water outlet header and oil filler integral. Block castings increase crank case rigidity and help maintain bearing alignment. Left block set 1¼ inches forward of the right to permit the lower end connecting rod bearings of opposite cylinders to be placed side by side on the same crank shaft pin to secure positive lubrication to the piston pin bearings and permit adjustment of the lower end connecting rod bearings. It also allows the use of a separate cam for each valve, eliminates the necessity for rockers and simplifies adjustment of valve tappets.

Ans. During the first part of the stroke, the temperature of the products of combustion is higher than that of the cylinder walls, hence, the gases are unduly cooled with a resulting fall of pressure in excess of that due to adiabatic expansion. During the latter part of the expansion (from *l* to *e′*, fig 2,689) the reverse conditions obtain, that is, the products of combustion absorb heat from the cylinder walls, resulting in an excess of pressure above that due to adiabatic expansion.

Pre-release.—In figure 2,689, it will be noticed that there is a sudden drop of the expansion line at *e′*. This is due to the opening of the exhaust valve before the end of the stroke or pre-release of the products of combustion to exhaust.

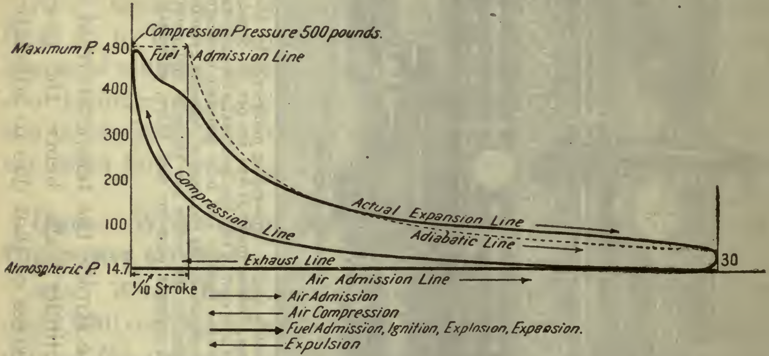


FIG. 2,692.—Indicator diagram of engine heating at constant temperature. For this condition of operation the final compression pressure should always be high enough to raise the final compression temperature up to the ignition temperature of the charge. The nearest actual approach to the cycle of an engine heating at constant temperature is that of the Diesel engine. The diagram represents the normal performance of a Diesel engine in which the temperature of the charge is raised to about 986° Fahr., under a compression of 500 pounds. It will be noted, that in this case the final compression temperature of 986° Fahr., not only represents the ignition temperature of the fuel used, but also the constant initial expansion temperature of the cycle. Since the heating is actually due to slow combustion rather than explosion, it is quite doubtful whether the temperature is really constant during the process.

Ques. Why does the exhaust valve open before the end of the stroke?

Ans. That the pressure may be reduced to a minimum at the beginning of the exhaust stroke.

Ques. Why does the pressure not fall abruptly at e', fig. 2,689?

Ans. Because an interval of time is required for the valve to open, during which, a corresponding movement of the piston takes place. The valve *begins* to open at e', presenting at first a very small opening for the exit of the burnt gases, this offers considerable resistance, which results in a gradual fall of pressure, as indicated by the line e' b'.

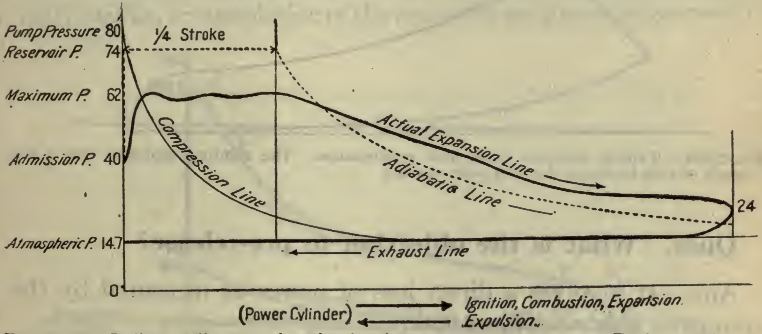


FIG. 2,693.—Indicator diagram of engine heating at constant pressure. Engine heating at constant pressure may operate by burning a charge introduced into the working cylinder under a compression pressure which will maintain the rate of inflow through the admission nozzle slightly greater than the rate of inflammation of the mixture, or by burning a charge, introduced into the working cylinder through a grating of wire gauze under a compression pressure which will supply a sufficient quantity of the charge to evolve enough heat, by slow but continued combustion, to maintain a given constant pressure in the cylinder during a part of the stroke of the piston. The nearest actual approach to the cycle of an engine heating at constant pressure is that of the Brayton engine. In the figure the charge is admitted at a compression pressure of 74 pounds and ignited the moment it passes through the grating. The heat evolved by the resulting combustion raises the pressure until it is equivalent to the constant initial expansion pressure required, and is maintained at that pressure by the continued admission of the charge during one-quarter of the piston stroke. The supply of mixture is then cut off, and the expansion allowed to continue until the pressure is reduced to very nearly that of the atmosphere at the end of the stroke. Although the cycle is that of a single acting engine, having one power stroke for every revolution of the crank, or two strokes of the piston, it corresponds more nearly to that of a four cycle engine than to that of a two cycle engine. It will be understood, of course, that the compression line represents the pressure conditions in the compression or auxiliary cylinder, which is one-half the size of the working cylinder. The fact, that compressing the charge in an auxiliary cylinder eliminates practically all the difficulties attending the act of compression in the working cylinder, and also permits the reduction of the final expansion pressure to very nearly that of the atmosphere without regard to the compression ratio, indicates that cycles heating at constant pressure ought to give more efficient engines than cycles heating at constant volume. The Brayton engine, however, never attained the success of any of the various types of engine heating at constant volume constructed previous to or since its time. The principal fault appears to have been due to poorly or improperly designed valves which caused an uneconomical consumption of fuel.

Ques. What other causes prevent an abrupt fall of pressure at pre-release?

Ans. A rich mixture with late ignition, especially at high speeds. Under these conditions combustion may not be complete at release, resulting in excess of pressure after the exhaust valve opens.

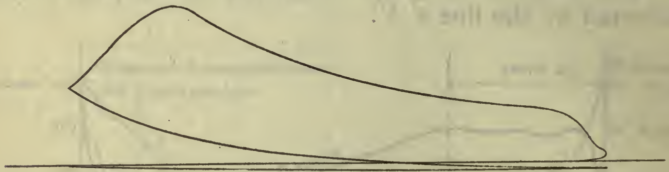


FIG. 2,694.—Typical indicator card, poor performance. The slanting explosion line is the result of slow explosion due to a poor mixture.

Ques. What is the objection to pre-release?

Ans. It involves a direct loss of power as measured by the triangular area *e e′ f*, fig. 2,696.

Ques. What is the objection to delaying the opening of the exhaust valve till the end of the stroke?

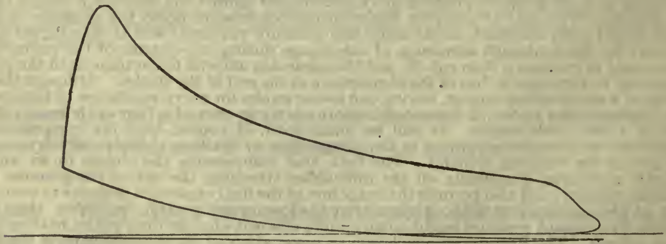


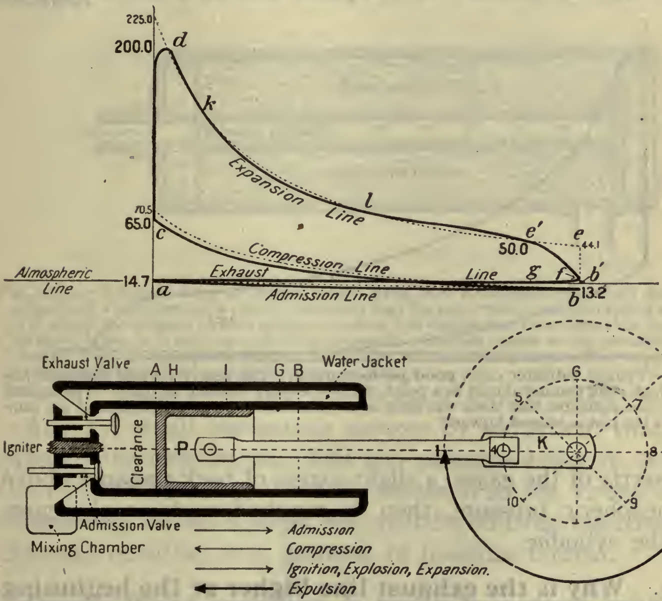
FIG. 2,695.—Typical indicator card; ***fair performance.*** This card represents the average workng conditions, the engine developing a moderate amount of power from a fairly strong mixture.

Ans. A similar, if not greater loss, would result by excess back pressure from b' to about g, fig. 2,696.

Ques. What is terminal pressure?

Ans. The pressure *that would exist* in the cylinder at the end of the stroke, if expansion were continued to this point.

Exhaust.—In figs. 2,696 and 2,697, which represent the exhaust stroke, or completion of the cycle, the backward movement of



FIGS. 2,696 and 2,697.—***Exhaust stroke.***—During this operation, the indicator pencil will trace a line *ga* which will either coincide with, or lie above, the atmospheric line according to the following conditions: 1. If the exhaust valve be properly set and there be no wire drawing the pressure will be reduced to that of the atmosphere when the piston reaches the point G, on its return stroke, and the exhaust line traced by the pencil will coincide with the atmospheric line. 2. If the exhaust valve be set so as to release at too high a pressure, and there be wire drawing, the pencil will trace a line which will lie above the atmospheric line at a distance corresponding to the resulting back pressure.

the piston from B, to A, expels the burnt gases from all parts of the cylinder except the clearance. During this operation, the indicator pencil will trace a line *ga*, which will lie more or less above the atmospheric line, especially at the beginning of the stroke.

Ques. Why does the exhaust line lie above the atmospheric line?

Ans. This is due to the frictional resistance which the outlet passages offer to the flow of the gases and in a small measure to the inertia of the gases; a slight excess of back pressure above the atmospheric pressure, then, is required to force the gases out of the cylinder.

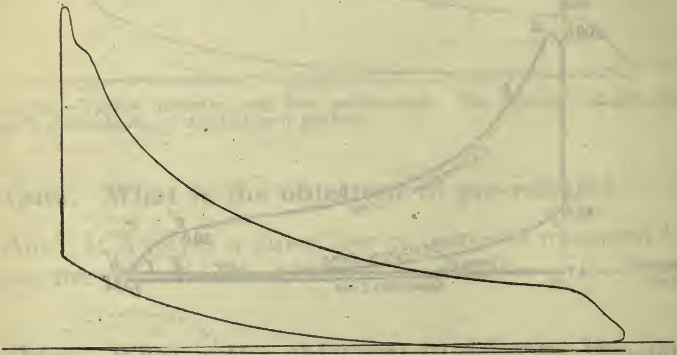


FIG. 2,698.—Typical indicator card; ***good performance.*** The nearly vertical explosion line with the high peak coming almost to a point, denotes a quick burning mixture. The smooth expansion line indicates that there was little or no after burning or continuation of the combustion as the piston moved forward.

Ques. Why is the exhaust line higher at the beginning of the stroke, as between *f* and *g*, fig. 2,696, than at other points of the stroke?

Ans. Generally there is not sufficient time during pre-release to exhaust the burnt gases to atmospheric pressure; the pressure gradually falls from f to g' being sustained in part by the inertia of the gases.

Ques. Does the movement of the piston during the exhaust stroke fully expel the gases?

Ans. No; only that portion displaced by its movement.

Ques. Does any part of the gases, remaining in the combustion chamber at the end of the exhaust stroke, escape?

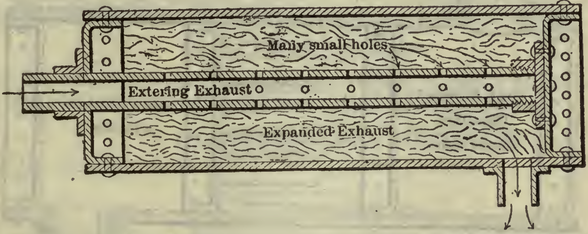


FIG. 2,699.—A simple form of muffler, which gives good satisfaction when well designed. It consists of a cylinder and a pipe so contrived that the pipe, which is drilled full of small holes, will admit the exhaust at high pressure, and as it is required to pass through a large number of small holes, it is split up and then expanded. The gas passes to the atmosphere in an uneven flow, at a pressure slightly above that of the atmosphere. This type of muffler is fairly efficient when well designed.

Ans. A small percentage escapes by induction, that is, the gases in the exhaust manifold and passages being in motion at the end of the stroke, tend to continue so by virtue of their momentum, hence, a slight flow is induced from the combustion chamber resulting in a decrease of pressure therein.

Ques. What provision is made to permit exhaust by induction at the end of the stroke?

Ans. The exhaust valve remains open a few degrees past the center, the exact amount depending, among other things, upon the speed of the engine.

Ques. What effect is caused by the pressure of non-combustible burned gases in the cylinder?

Ans. This is a fertile source of lost efficiency, as is proved by the superior average performance of scavenging engines, in which these gases are largely expelled.

A mixture of 9 to 1, with no burned gases present, gives a rise of about 2,373 degrees; the same mixture, compressed with the burned gases of a previous explosion in a clearance of 41⅔ per cent of the cylinder volume gives a rise of only about 1,843 degrees.

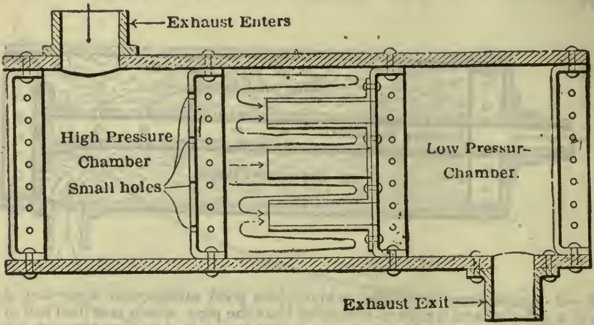


FIG. 2,700.—Three chamber type of muffler: a high pressure, intermediate, and low pressure chamber, so contrived that the pressure is reduced almost to zero before the exhaust makes its exit to the atmosphere. The volume of the high pressure chamber is equal to that of one of the engine cylinders; the intermediate chamber has twice the volume, and the low pressure chamber three times the volume of the engine cylinder.

Ques. What feature of construction causes a considerable amount of back pressure?

Ans. The muffler; its numerous compartments and tortuous passages all offer frictional resistance to exhaust.

A French authority states that an engine of 8.1 H. P., running with muffler, gave 6.1 B. H. P. at 967 revolutions per minute, but with muffler gave the same power only on 1,012 revolutions. He also found for a 2.25 I. H. P. an efficient output of 2.16 at 2,015 revolutions without muffler, and of 1.91, at 2,057 revolutions with muffler, claiming a loss of 145 foot pounds per second.

These figures are very typical for many mufflers, and, although possibly reduced by some of the more modern models, represent fairly well the kind of obstacles obtruded in the way of the highest mechanical efficiency of the average gas engine.

Ques. What takes place during exhaust?

Ans. A large amount of heat and power units are lost.

Ques. Why cannot these losses be avoided?

Ans. It is not practical to expand the gas, after explosion, to atmospheric pressure within the cylinder.

For instance, if the terminal pressure of expansion be 45 lbs. absolute, in order to expand this to atmospheric pressure would require

$$45 \div 14.7 = 3.06 \text{ expansions}$$

To secure this additional expansion, an engine, of, say 5 inch stroke, and 40 per cent clearance would require an increase of stroke equal to

$$\{ (40 \text{ per cent of } 5) + 5 \} \times 3.06 = 21.4 \text{ ins.}$$

making the total stroke

$$21.4 + 5 = 26.4 \text{ ins.}$$

It is readily seen that such abnormal proportions are not practical, either for working conditions or for efficiency; in the latter case, the gain due to the increased expansion would be largely offset by the extra friction due to the long stroke, and other conditions causing loss.

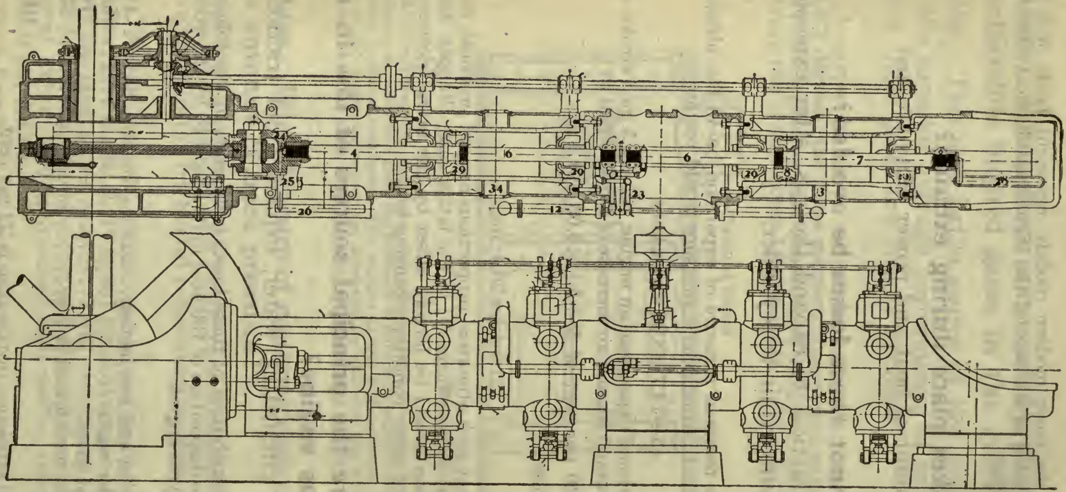
Ques. What are the principal sources of loss in the operation of a gas engine?

Ans. Radiation, circulating water, exhaust, and friction.

Some idea of the distribution of these losses is afforded by the following table, which shows the disposition of 100 heat units absorbed by the working substance of a four cycle engine.

Heat lost by radiation and conduction	11
Heat lost to cylinder walls	53
Heat lost in exhaust gases	14
Heat converted into mechanical energy	22
	100

The actual thermal efficiency in this case is 22 per cent.



FIGS. 2,701 and 2,702.—Plan and elevation of National horizontal tandem four cycle double acting engine. Each cylinder with its jacket is a single casting. The jacket walls are open around the center of the cylinder and are closed by a four piece cast iron band acting as an expansion joint. The piston and rods are cored and bored (respectively) for circulation of cooling water. The inlet valves are located in the top of the cylinders and the exhaust valves in the bottom. The valves operate in cages arranged to permit removal without disturbing any of the piping. Each inlet valve and its respective exhaust valve are operated by means of rockers and wiper cams actuated by valve rods from a common eccentric. The lay shaft, operating these eccentrics, is spur gear driven from the main crank shaft. Regulating is by a Jahre governor which operates the mixing valves, controlling the speed by throttling. The mixing valves are hand adjustable, for proportions of gas and air, and are located in the same case as the inlet valves. An automatic safety stop shuts down the engine by tripping a switch in the ignition circuit in case the governor fail to control the speed. Make and break ignition is used, operated from cams on the lay shaft. Each igniter has an independent timer. Each combustion chamber is equipped with an air starting valve actuated by cams on the main lay shaft.

The accompanying table has been prepared for obtaining the usual pressures and temperatures of exhaust corresponding to various degrees of expansion.

Table of Pressure and Temperature Coefficients

(Terminal)

E	P	T	E	P	T
3.	23.975	719.22	4.	16.494	659.75
3.05	23.464	715.66	4.1	15.973	654.88
3.1	22.973	712.18	4.2	15.480	650.17
3.15	22.5	708.75	4.3	15.014	645.55
3.2	22.045	705.43	4.4	14.572	641.16
3.25	21.605	702.15	4.5	14.152	636.85
3.3	21.180	698.95	4.6	13.745	632.66
3.35	20.770	695.80	4.7	13.374	628.60
3.4	20.374	692.72	4.8	13.013	624.63
3.45	19.991	689.68	4.9	12.669	620.8
3.5	19.621	686.72	5.	12.340	617.03
3.55	19.262	683.80	5.1	12.027	613.37
3.6	18.915	680.94	5.2	11.727	609.81
3.65	18.580	678.14	5.3	11.440	606.34
3.7	18.253	675.36	5.4	11.166	602.95
3.75	17.937	672.66	5.5	10.903	599.64
3.8	17.631	669.99	5.6	10.650	596.41
3.85	17.334	667.36	5.8	10.175	590.15
3.9	17.046	664.79	6.	9.737	584.19

To ascertain the exhaust pressure corresponding to a given explosion pressure, and given explosion pressure is multiplied by the number in the column marked P, opposite the number in the column marked E corresponding to the given expansion ratio, and the product divided by 100. The result is the absolute exhaust pressure required.

Example.—If the explosion pressure be 225 lbs., and the expansion ratio 3.8, what is the possible exhaust pressure?

Applying the rule to the table

$$\text{Probable exhaust pressure} = \frac{(225+14.7)\times 17.631}{100} = 42.3 \text{ lbs. absolute}$$

or $42.3 - 14.7 = 27.6$ lbs. gauge pressure.

To ascertain the exhaust temperature corresponding to a given explosion temperature, the given explosion temperature is multiplied by the number in the column marked T, opposite the number in the column marked E, corresponding to the given expansion ratio, and the product divided by 1,000.

Example.—If the explosion temperature be 2,300° Fahr., and the ratio of explosion 3.8, what is the probable exhaust temperature applying the rule to the table?

Probable exhaust temperature = (2,300 + 459.2) × 669.99 = 1,848.5 absolute or 1,848.5 − 459.2 = 1,389.3° Fahr.

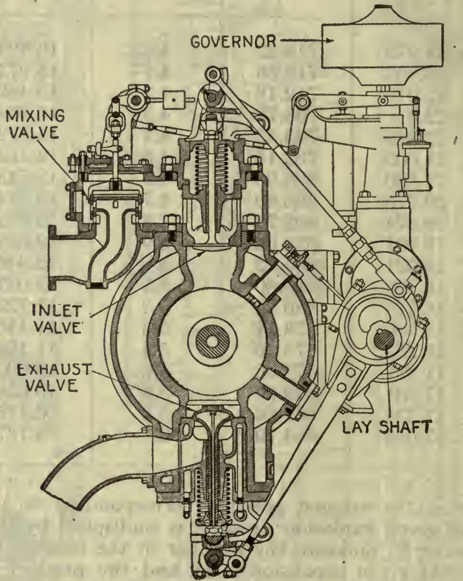


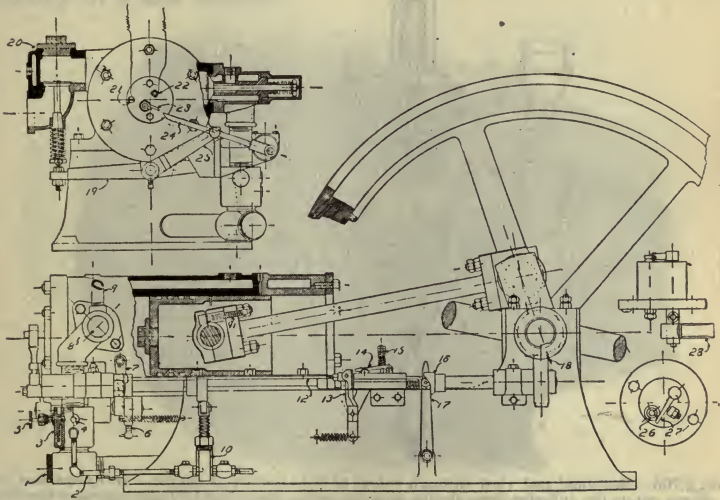
Fig. 2,703.—Sectional view of National horizontal tandem four cycle double acting engine showing mixing, inlet, and exhaust valves, valve gear and governor.

Ques. What is understood by the term "exhaust temperature" as here used?

Ans. The temperature of the gases at the terminal pressure. The temperature, of course, decreases as the gases expand to the lower pressure of the exhaust stroke.

Efficiency.—The theoretical thermal efficiency of an engine depends upon the temperature range, that is, the difference between the initial and final temperature of the working substance.

If the temperature range be increased, either by raising its upper limit or by decreasing its lower limit, or by the combined increase of the one and decrease of the other, the result will be an increase in the amount of heat available for conversion into mechanical energy, and consequently an increase of thermal efficiency of the engine.



FIGS. 2,704 and 2,705.—End and longitudinal sectional views of Bauroth horizontal four cycle engine. ***The parts are:*** 1, air plug; 2, gasoline feed pipe; 3, carburettor; 4, gasoline overflow; 5, needle valve; 6, exhaust relief screw; 7, exhaust relief lever; 8, inlet valve; 9, gas inlet; 10, gasoline lift pump; 11, adjusting screw; 12, exhaust relief rod; 13, governor hit and miss lever; 14, governor pawl; 15, speed regulating spring; 16, governor cam; 17, governor cam lever; 18, secondary gears; 19, exhaust lever; 20, exhaust box; 21, post for wire; 22, stationary electrode; 23, igniter pawl; 24, igniter rod; 25, igniter control lever; 26, stationary electrode; 27, movable electrode; 28, igniter trip pawl.

Ques. Define the theoretical thermal efficiency.

Ans. It is the difference of the initial and final temperatures divided by the initial temperature, that is,

$$\text{theoretical thermal efficiency} = \frac{T' - T''}{T'} \ldots\ldots (1) \text{ in which}$$

T' = initial temperature absolute
T'' = final temperature absolute

Example.—What is the theoretical thermal efficiency of a gas engine in which the initial temperature is 2,800° and the final temperature 80° Fahr.?

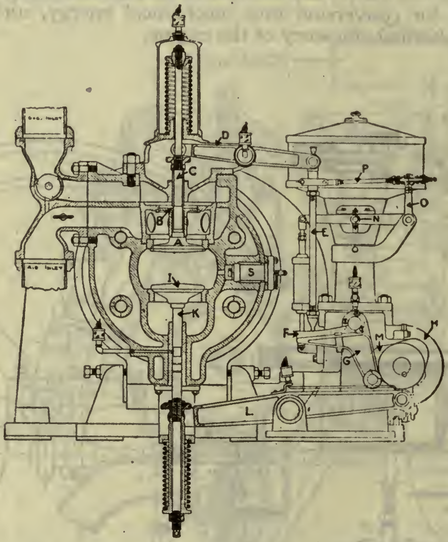


FIG. 2,706.—Sectional end view through valves of Otto four cycle engine. ***In operation,*** the charge enters the cylinder through two separate ports, controlled by hand operated throttles, and the inlet valve poppet A. The poppet B, mounted flexibly on the valve stem C, controls the admission of the gas in relation to the air inlet. The inlet cam H, acting upon the lever G, upon which the steel shoe F, is mounted, opens the inlet valve A, in a positive manner by means of the push rod E, and the valve lever D. The exhaust gases are expelled through the poppet valve I-K, which is lifted and timed by the lever L, and the exhaust cam M. A powerful centrifugal governor, driven by spiral gears from the cam shaft, regulates the volume of the charge admitted to the cylinder in accordance with the variations in the load carried by the engine. To accomplish this, the lift of the inlet valve A, is decreased or increased when the pivot pin N, is raised or lowered by the governor, thus causing the lower end of the push rod E, to be moved by means of lever O, and rod P, closer to or farther away from the fulcrum of the lever G. It will be readily understood that as there is always a slight amount of play between the end of the rod E, and the steel shoe F, the work performed by the governor in swinging the rod E, is very light and reduces the stress and wear on the governor parts to a minimum. A dash pot consisting of a cylinder and piston, oil cushioned, and connected to the governor lever, steadies the operation of the governor and prevents sudden variations of speed under fluctuating loads. S, represents the igniter.

In the case of the gas engine, the initial temperature means the explosion temperature, and the final temperature, that of the exhaust. Reducing the given values to the absolute scale.

$$T' = 2{,}800 + 459.2 = 3{,}259.2°$$
$$T'' = 80 + 459.2 = 539.2°$$

substituting in (1)

$$\text{Theoretical thermal efficiency} \times \frac{3{,}259.2 - 539.2}{3{,}259.2} \times 83 \text{ per cent}$$

Ques. What may be said of the actual thermal efficiency of a gas engine?

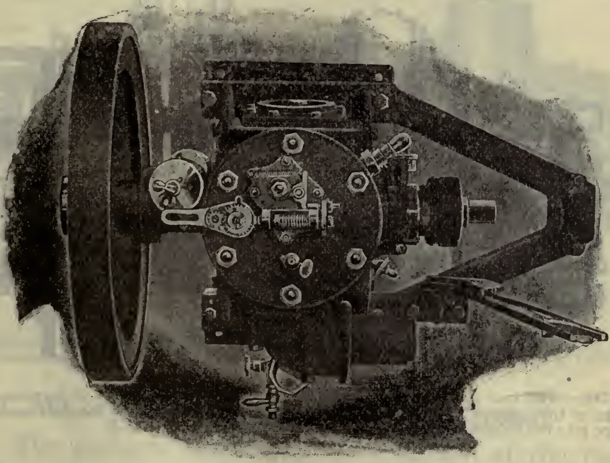


FIG. 2,707.—Top view of Fay and Bowen two cycle marine engine.

Ans. For gasoline engines, in general, the actual thermal efficiency ranges from almost 10 to 20 per cent.

The large margin between the theoretical and actual thermal efficiency of a gas engine is accounted for in heat balance given in the preceding table; the various losses indicated in the table should be carefully noted.

Two Cycle Engines.—This type of gas engine is used chiefly for small powers, especially for marine service. The essential difference between it and the four cycle type is that the four operations of admission, compression, impulse, and exhaust, comprising the working cycle, are performed in one revolution instead of two. There is then, one impulse for each revolution. From this, it follows that the weight is much less than that required for the four cycle engine. The necessary mechanical features for two cycle operation are:

1. An enclosed crank case fitted with a valve arranged to open and admit fuel gas at the front, instead of at the rear of the piston, on the inward, instead of the outward stroke, as in the four part cycle.

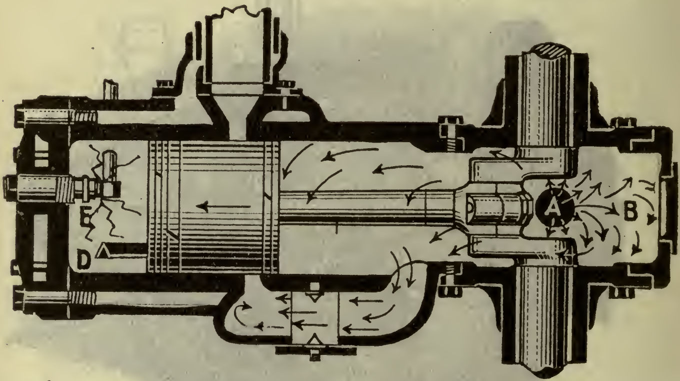


FIG. 2,708.—The two cycle engine: ***first stroke.*** The inward stroke of the piston induces a charge of the mixture at A into the crank case B, and compresses the previously admitted charge into the cylinder D; the subsequent ignition takes place at E.

2. Inlet and exhaust ports located at points near the extreme outward position of the piston, so as to be uncovered during the outward stroke.

3. A by pass tube connecting the interior of the cylinder with the crank case, so as to admit fuel gas at the proper point in the cycle.

Ques. In the operation of a two cycle engine, what occurs during the first stroke?

Ans. The piston moves inward and draws in a charge of the explosive mixture into the enclosed crank case. During the operation the charge previously admitted to the cylinder is compressed and ignited as the piston nears the end of the stroke as in fig. 2,708.

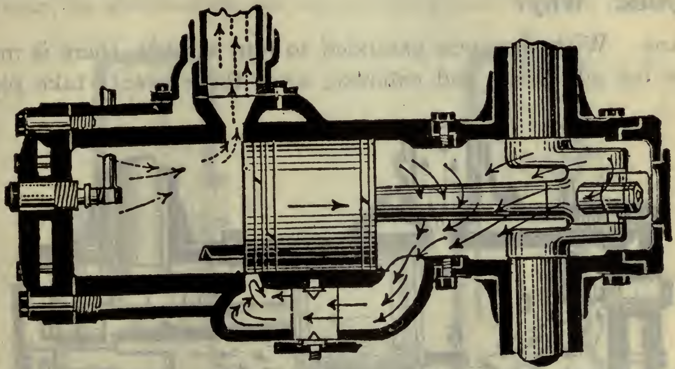


FIG. 2,709.—The two cycle engine: *first part of* **second stroke.** The outward stroke of the piston uncovers the exhaust port, thus releasing the burnt gases in the cylinder, and simultaneously compressing the previously admitted charge in the crank case.

Ques. What occurs during the second stroke?

Ans. The pressure caused by the explosion of the charge drives the piston outward, and slightly compresses the mixture drawn into the crank case during the previous stroke as in fig. 2,709. Near the end of the stroke the piston uncovers the exhaust port and the burnt gases are exhausted. During the remainder of the stroke the piston uncovers the admission port, as in fig. 2,710, and the new charge, previously compressed in the crank case, is admitted to the cylinder, being deflected upward to the head end of the cylinder by a screen or "deflector plate" set in the end of the piston.

The "inrush" of the new charge helps materially to clear the cylinder of the burnt gases from the previous charge. The object of the deflector plate is to prevent the entering charge passing out through the exhaust in place of the burnt gases.

Ques. Which type of engine is the more economical?

Ans. The four cycle engine.

Ques. Why?

Ans. With the cycle extended to four strokes, there is more time for admission and exhaust; since these events take place at separate intervals, no chance is given for any of the charge to escape past the exhaust valve while open. Owing to simultaneous admission and exhaust in the two cycle engine, **pre-release** of the burnt gases must take place earlier than in the four cycle engine, resulting in a loss of power which is avoided in the latter. The inefficiency of admission and exhaust of the two cycle engine becomes more marked at high speeds.

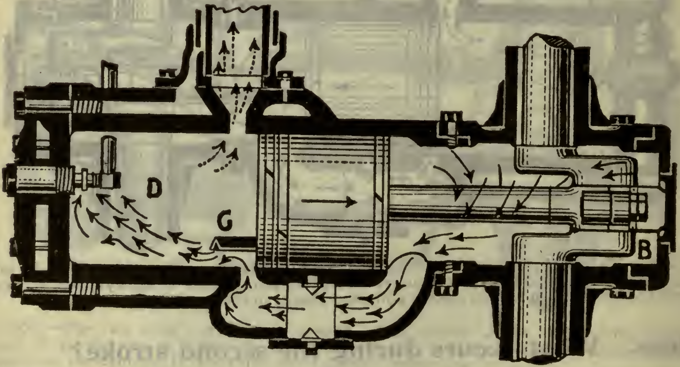
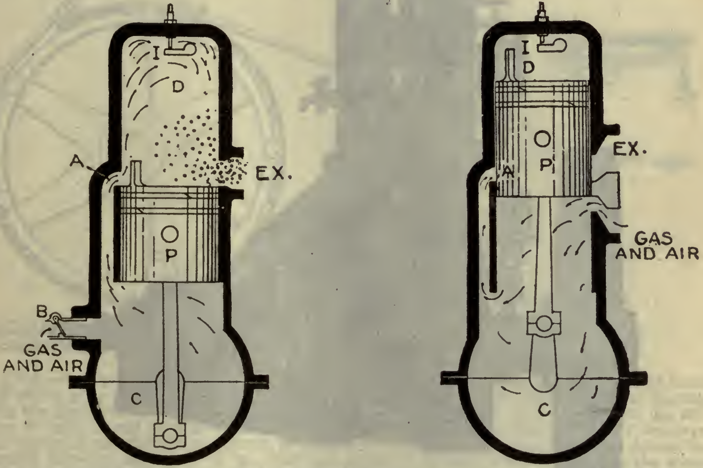


FIG. 2,710.—The two cycle engine: *end of the* ***second or outward stroke.*** The gases compressed in the crank case are admitted to the cylinder space, D through the open inlet port, and sometimes past the screen or deflector, C. The passage between the cylinder and the crank case is controlled by a butterfly valve, which here, is shown open.

The successful operation of the two cycle engine at high speeds will depend on adequate provision for rapid exhaust. A prominent gas engine authority remarks:

"The two cycle engine, at best, is the next thing to an impossibility." By this statement, he means that the act of admitting inflammable fuel mixture into the cylinder, already filled with flaming gas, without igniting it, involves something closely approaching a contradiction in physical conditions.

Were it not for the fact that the burning gases actually exhaust faster than the new mixture is admitted under impulse of their inherent expansion, the ignition of the new charge would seem to be nearly inevitable.



Figs. 2,711 and 2,712.—Sectional views of Palmer two cycle engine showing difference between the ***two port*** and ***three port*** types. Fig. 2,711, two port engine; fig. 2,712, three port engine. In the three port engine when the piston is at top of stroke a vacuum is created in base (see fig. 2,712). Inlet port, marked *gas and air*, is opened, allowing a charge to enter crank case. When piston starts down, this port is closed. The mixture of gas and air in crank case C, is then compressed by the downward motion of piston. Transfer point A, is opened, allowing gas and air which has been mixed in crank case to pass into cylinder. On the downward or explosion stroke the exhaust port opens slightly before transfer port and the exhaust or burned gases are passing out as a fresh charge is coming in through transfer port. To prevent this incoming charge passing across piston and into exhaust port a baffle plate D, is placed on top of piston which deflects a charge of incoming gas to top of cylinder. Piston comes up again closing exhaust port and transfer port and mixture is exploded. In the two port type, fig. 2,711, a check valve is used in place of the third port as shown.

By deflecting the incoming mixture to the rear end of the cylinder, it follows the rapidly expanding exhaust, coming into contact with it only when the expansion has so far reduced the temperature that the danger of pre-ignition is averted.

Ques. For what service is the two cycle engine extensively used?

Ans. For marine use, especially to propel boats of small size.

Ques. Why is this?

Ans. On account of its light weight, simplicity, and small space, as well as low cost of construction.

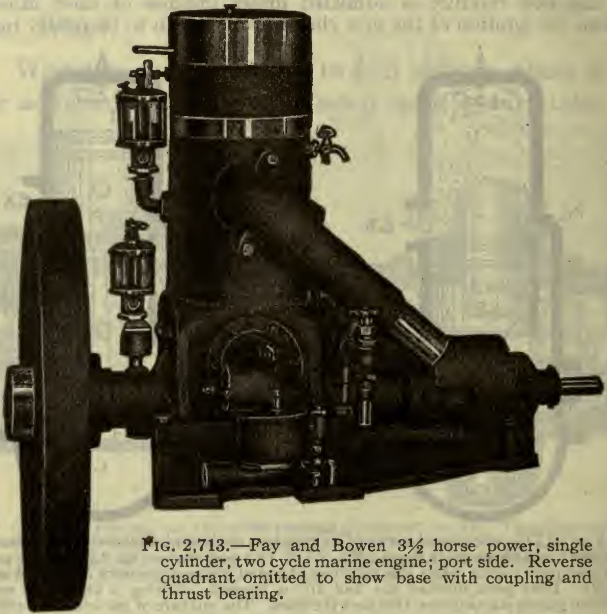


FIG. 2,713.—Fay and Bowen 3½ horse power, single cylinder, two cycle marine engine; port side. Reverse quadrant omitted to show base with coupling and thrust bearing.

Horse Power of Gas Engines.—The term horse power, correctly used is absolute, meaning one and only one thing, but in connection with the automobiles and motor boats it has been so loosely used that its utility has largely been lost.

Ques. What is generally understood by the term horse power in speaking of gas engines?

Ans. The brake horse power.

Ques. What is brake horse power?

Ans. The actual horse power delivered by the engine; the word brake is applied because the power is determined by a brake test as shown in fig. 2,714.

Ques. Can the brake horse power be approximated from the indicated horse power?

Ans. Yes, by making suitable allowance for friction.

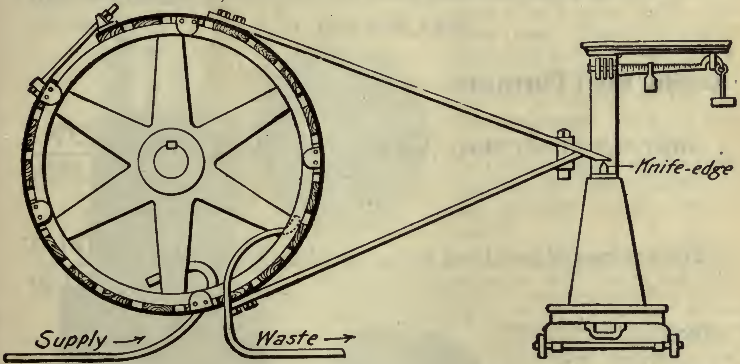


FIG. 2,714.—Prony brake for determining brake horse power. It consists of a friction band ring which may be placed around a pulley or fly wheel, and attached to a lever bearing upon the platform of a weighing scale in such a manner that the friction between the surfaces in contact will tend to rotate the arm in the direction in which the shaft revolves. This thrust is resisted and measured *in pounds* by the scale. In setting up the brake the distance between the center of the shaft and point of contact (knife edge) with the scales must be accurately measured, *the knife edge being placed at the same elevation as the center of the shaft.* An internal channel permits the circulation of water around the interior of the rim as shown, to prevent overheating.

Ques. How much allowance should be made for friction?

Ans. The mechanical efficiency varies greatly, depending upon the design, workmanship, and type of engine, although no general rule can be given, yet in well designed automobile engines which are operated near the speed at which they are designed to give the best results, the mechanical efficiency may be taken as varying between 80 and 90%. Below 80 is not good; 90 or above is exceptionally good, and for engines of automobile type, from 15 H.P., upward, the efficiency can be assumed as about 85%.

Example.—If the indicated horse power of an engine be 58.8, what is the calculated brake horse power for a mechanical efficiency of 85 per cent?

In this case, 15 per cent of the power developed in the cylinder was spent in overcoming friction, hence, the actual power developed at the shaft is

$$58.8 \times .85 = 50 \text{ H. P.}$$

Racing Boat Formulæ.—

American Power Boat Association, horse power $= \dfrac{D^2N}{2.5338}$

For engines of less than 6 ins. stroke, horse power $= \dfrac{D^2L\,N}{15.20}$

in which

D^2 = square of piston diameter in inches;
L = stroke in inches;
R = revolutions per minute;
N = number of cylinders.

These formulæ are for high speed racing boat engines of the four cycle type, and are based on 1,000 feet piston speed per minute.

For engines of ordinary design, about ⅔ of the above values should be taken. 10 per cent should be added to the ratings if the charge is forced into the cylinders by any mechanical device.

Two Cycle Horse Power Formulæ.—The American Power Boat Association gives the following for two cycle engines:

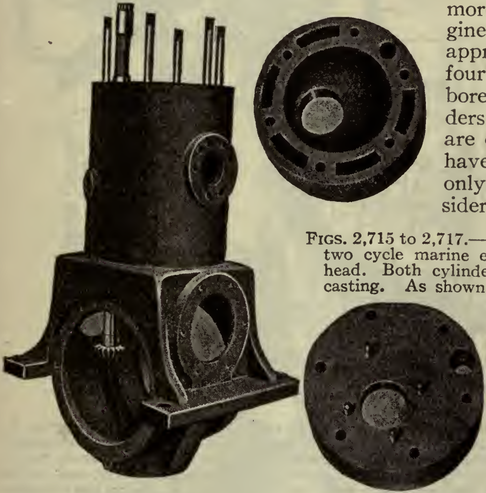
$$\text{horse power} = \frac{D^2\,N}{2.1008}.$$

$$\textit{horse power} = \frac{D^2 N}{12.987}$$

These formulæ are only for racing boat engines. For ordinary two cycle engines about ⅔ of the value obtained in the above should be taken.

For engines having one or more displacer cylinders, the above rating should be increased in the ratio that the displacer piston's displacement bears to that of the working cylinders.

It will be noticed that a 2 cycle engine having double the number of power strokes of a four cycle, the H. P. would be multiplied by 2. This, however, would give an erroneous result, as there are many inherent conditions connected with 2 cycle engine design which tend to lower its H. P. output, such as the lower compression, lower M. E. P., due to inefficient scavenging, etc. For these reasons the output of a 2 cycle engine varies more than it does in 4 cycle engines, and is very often taken as approximately 1.65 of that of a four cycle engine of the same bore and stroke. It must be understood, however, that there are cases where 2 cycle engines have shown better results, but only average results are considered.



FIGS. 2,715 to 2,717.—Fay and Bowen, 3½ horse power, two cycle marine engine main casting and cylinder head. Both cylinder frame and bed plate form one casting. As shown, the driving shaft of the guide passes up through the main casting. Where it passes through the water jacket, this ignitor shaft is cased in a water tight tube. The bevel pinion at the lower end of the ignitor shaft meshes into a bevel gear attached to the crank shaft. The cylinder head has a hole near the center. This is where the igniter plug is inserted. This plug is "ground in," and held in place by nuts on the four studs. If it should be desirable at any time to examine the cylinder without removing the cylinder head, this can be done by taking out the igniter plug, as the hole is large enough to expose to view a large part of the interior.

Parts of a Gas Engine.—A gas engine is composed of stationary and moving parts; the stationary parts are: 1, the cylinder, and 2, crank case; the moving parts are: 1, the piston, 2, connecting rod, 3, shaft, 4, fly wheel and 5, valve gear.

Ques. Describe the cylinder?

Ans. The cylinder is open at the end toward the crank and as varying between 80 and 90%. Below 80 is not good; 90 or above is exceptionally good, and for engines of automobile type, from 15 H.P., upward, the efficiency can be assumed as about 85%.

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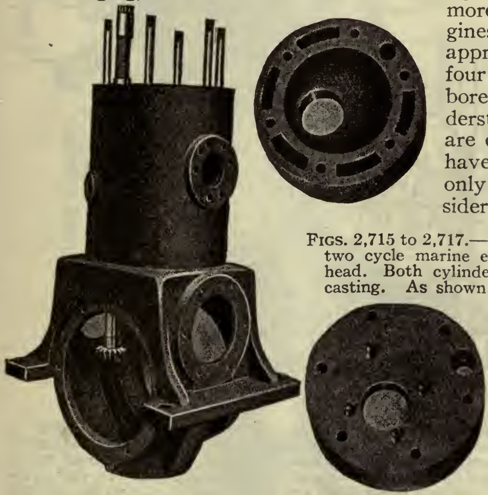


FIGS. 2,715 to 2,717.—Fay and Bowen, 3½ horse power, two cycle marine engine main casting and cylinder head. Both cylinder frame and bed plate form one casting. As shown, the driving shaft of the guide passes up through the main casting. Where it passes through the water jacket, this ignitor shaft is cased in a water tight tube. The bevel pinion at the lower end of the ignitor shaft meshes into a bevel gear attached to the crank shaft. The cylinder head has a hole near the center. This is where the igniter plug is inserted. This plug is "ground in," and held in place by nuts on the four studs. If it should be desirable at any time to examine the cylinder without removing the cylinder head, this can be done by taking out the igniter plug, as the hole is large enough to expose to view a large part of the interior.

Parts of a Gas Engine.—A gas engine is composed of stationary and moving parts; the stationary parts are: 1, the cylinder, and 2, crank case; the moving parts are: 1, the piston, 2, connecting rod, 3, shaft, 4, fly wheel and 5, valve gear.

Ques. Describe the cylinder?

Ans. The cylinder is open at the end toward the crank and closed at the opposite end save for inlet and exhaust ports, which are opened and closed by valves. The cylinder is surrounded by an annular space of jacket through which cooling water circulates.

Ques. What is the combustion chamber?

Ans. This corresponds to the clearance space in a steam engine cylinder, but its object is to provide a small space into which each charge of the fuel mixture may be compressed to considerable pressure on the compression stroke of the piston, and then ignited. Together with the volume displaced by the piston, it forms the total content of the cylinder as measured when the piston has reached the end of the *outward* stroke.

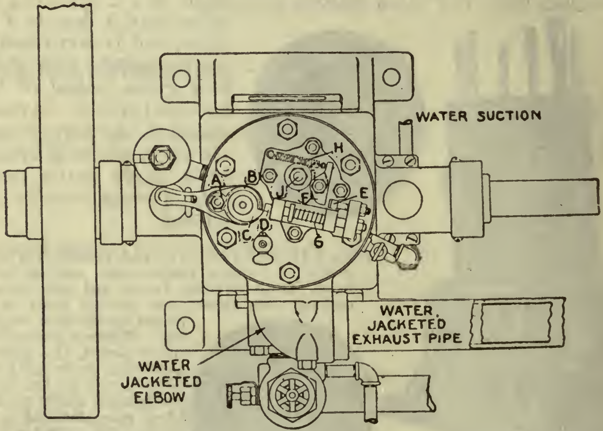


FIG. 2,718—Plan of Fay and Bowen two cycle engine. ***In construction,*** the smaller gear A, is attached to the ignitor shaft, and turns with it. The gear drives a larger gear B, the teeth of which show around and below the cam C. ***In operation,*** as the ignitor shaft turns, this larger gear B, turns the cam C. The cam C, pushes back the plunger D, and as the plunger D, slips off the point of the cam C, its other end or hammer E, strikes the movable electrode F, breaking the electric current, and producing a spark. In the line drawing, the relation of the electrodes and sparking points *inside* of the firing chamber is indicated by dotted lines, the spring H, brings the movable electrode F, to its place again, when the plunger D, backs off for the stroke.

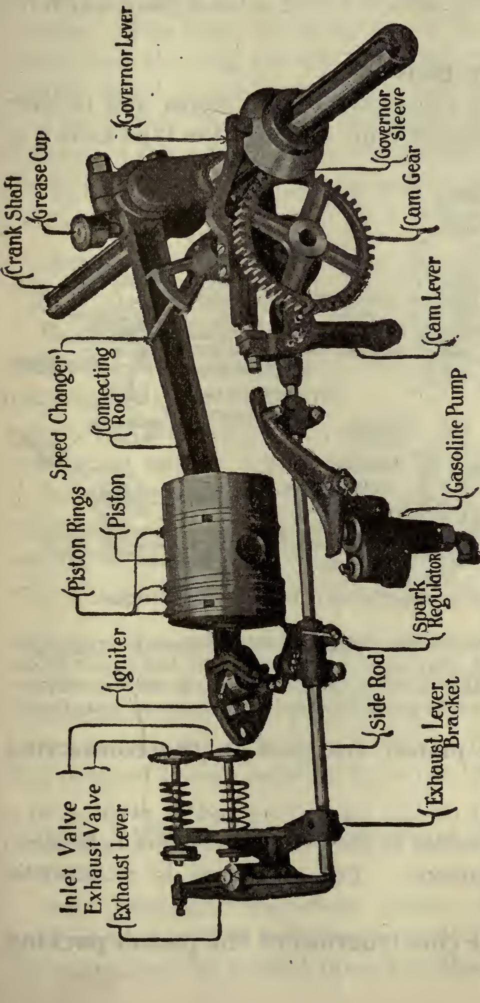


FIG. 2,719.—Working parts of Alamo four cycle engine.

Ques. What other name is given to the combustion chamber?

Ans. It is sometimes called the **clearance,** as it is the space which is not displaced by the piston on its inward stroke.

Ques. What does the size of the combustion chamber determine?

Ans. The degree of compression; the smaller the chamber in proportion to the piston displacement, the higher the pressure to which the mixture is compressed.

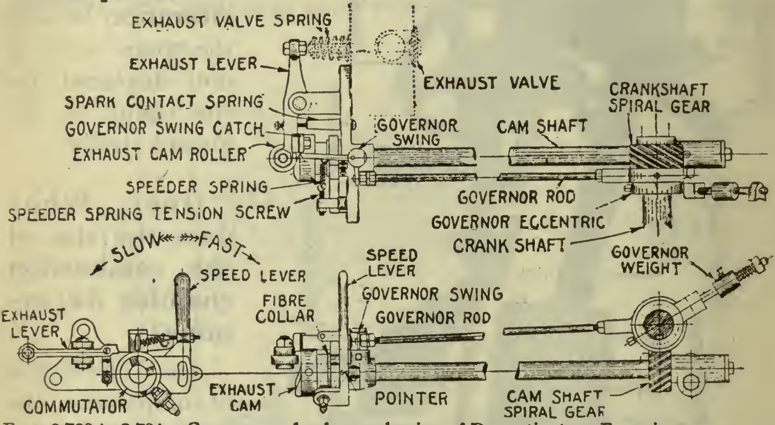
Ques. How is the cylinder supported?

Ans. By the crank case, which

is a closed box-like structure made up of an intermediate piece to which the cylinder is attached, and a lower piece which retains the oil.

Ques. Describe the piston.

Ans. It consists of a cylindrical box of proper size to slide back and forth in the cylinder bore, and is of the type known as "trunk piston."



FIGS. 2,720 to 2,724.—Governor and valve mechanism of Domestic, type F, engine.

This form of piston performs the duties of both piston and cross head. The piston is single acting, that is, it is acted upon by the power on one face only. To prevent leakage, several grooves are cut in its circumference to receive packing rings, the type generally used being known as "snap rings."

Ques. How is the piston attached to the connecting rod?

Ans. The upper end of the connecting rod is pivoted to a "wrist pin," which is inserted in the piston through a hole bored through its central diameter. The wrist pin is retained in place by two set screws.

Ques. Describe the construction of the piston packing rings.

Ans. They are made from a pipe shaped casting, which is turned in a lathe to an outer diameter slightly larger than the cylinder. The inner circumference is then turned from another center, and the ring cut off and split at the thinnest section. Enough metal is removed at the split section so that the ring may be slightly compressed and again turned externally to the diameter of the cylinder while compressed. The latter operation causes the ring to bear evenly against the cylinder walls, thus making a tight joint.

Although formed of a brittle substance, piston rings have considerable elasticity, being capable of opening sufficiently to be slid over the outer diameter of the piston, and to "snap" back into the grooves.

Ques. What is the use and construction of the connecting rod?

Ans. It transforms the to and fro motion of the piston into rotary motion and transmits the thrust or power impulse of the piston to the shaft. The connecting rod is usually of rectangular or I shaped construction, and has an adjustable bearing at each end. One end is pivoted to the piston by the wrist pin, and the other end to the shaft by the crank pin.

Ques. Describe the crank shaft.

Ans. This is formed from a solid steel forging, and consists of a central cylindrical piece, from which radiate one or more center cranks, corresponding to the number of connecting rods. The order of cranks depends upon the type of the engine.

Ques. What kind of valves are used on a four cycle engine?

Ans. Poppet or mushroom valves. These consist of metal discs beveled around one face, so as to fit into a countersink in the port; they are carried upon spindles.

Ques. How many valves are necessary?

Ans. An admission or inlet valve, and an exhaust valve for each cylinder.

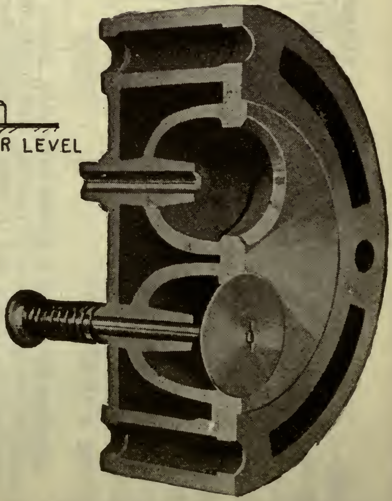


Fig. 2,725.—Sectional view of Dempster cylinder head showing valve construction.

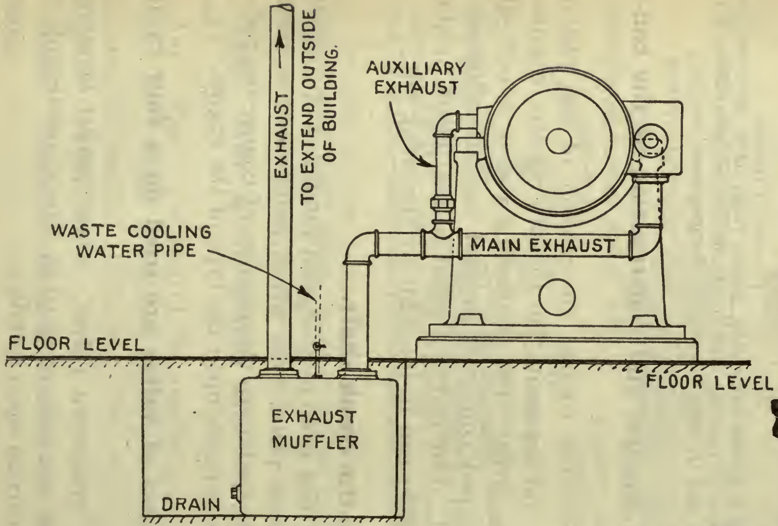


Fig. 2,726.—Arrangement of exhaust piping showing DuBois auxiliary exhaust, designed to prevent overheating of the valves.

Ques. Name the two types of admission valve.

Ans. The automatic, and the positive or mechanically operated valve.

The automatic inlet valve, operated by suction of the piston against the tension of a spiral spring, has been extensively used, but has been largely replaced by the positive type. The reasons for this change are: 1, that the automatic valve often sticks with gummed oil on its seat; 2, that the spring tension may vary, thus changing the fuel pressure in the cylinder; 3, that it is noisy; 4, that its operation on high speed engines is unreliable. As against these defects, the positive inlet valve possesses the advantages of opening and closing as desired without noise or sticking, and of giving the same amount of opening at both high and low speeds. The exhaust valve is always operated mechanically.

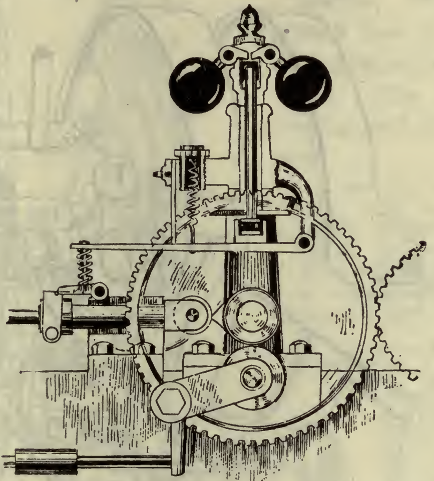


FIG. 2,727.—Du Bois governor. ***In operation,*** when the engine reaches a speed above normal the governor forces a latch down into an adjustable stop collar, which holds the exhaust valve open, and immediately shuts off all fuel supply, allowing the gas and exhaust valves to remain idle and the piston to travel in the cylinder without compressing idle charges of air. The engine has two exhausts: main, and auxiliary. The auxiliary exhaust is an opening called a port on the lower left side of the cylinder. This opening or port is not uncovered until the charge drawn into the cylinder has expended its force in power and the piston has completed its expansion or working stroke. When the auxiliary exhaust port is uncovered, a large part of the exploded gases escape to the muffler, the remaining dead gases being scavenged through the main exhaust valve.

Ques. Describe the valve gear.

Ans. The valve stem is made of sufficient length to extend down into the crank case; it enters through a bushing which serves as a guide. Attached to the end of the stem is a roller bearing, which rides on a cam attached to the cam shaft. The latter is geared to the crank shaft in such proportion that it makes one revolution to every two of the crank shaft. By means of a spring, the roller bearing is held in contact with the cam.

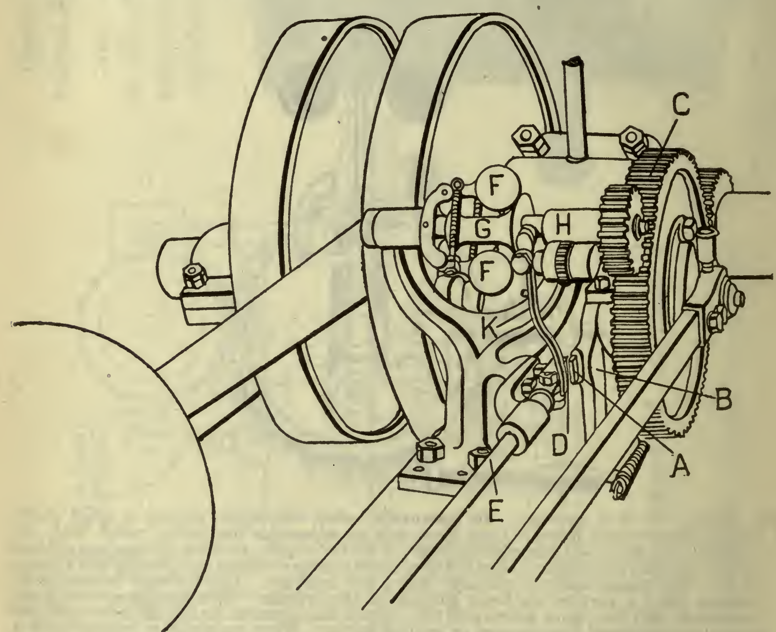


FIG. 2,728.—Foos hit or miss governor. ***In operation,*** at normal speed, the blade A, on the fuel lever B, operated by a cam on the gear wheel C, strikes the notched finger D, on the end of the push rod E, causing the fuel valve to open and admit a charge into the cylinder at each revolution of the cam. As the speed increases, the governor balls F, move outward, causing the spool G, to push against the roller H, carried on the upper end of the governor lever K. This action causes the lower end of the lever to move inward and push the notched finger D, out of its normal position, so that the fuel lever blade A, fails to hit it, and thus allows the fuel valve to remain closed. The fuel supply being thus cut off, the speed decreases and the governor balls return to their original position, releasing the pressure on the lower end of the governor lever and allowing the notched finger D, to resume its regular position under the pressure of a spring. It will be understood in this connection, that centrifugal fly ball governors operating on the hit or miss principle may be attached horizontally to a transverse secondary shaft as in the foregoing case, or they may be placed vertically at the side of the cylinder and receive their motion from a longitudinal secondary shaft through the medium of bevel gearing.

Ques. Are two cam shafts necessary?

Ans. The valves may be operated from a single cam shaft when they are located on one side of the cylinder; when the valves are on opposite sides, two cam shafts are needed.

Ques. What is necessary for the operation of a gas engine?

Ans. The fuel, ignition, cooling, and lubrication systems.

A defect in any of these will affect the operation of the engine, hence it is desirable to clearly understand the principles involved in the working of these several systems, as later explained.

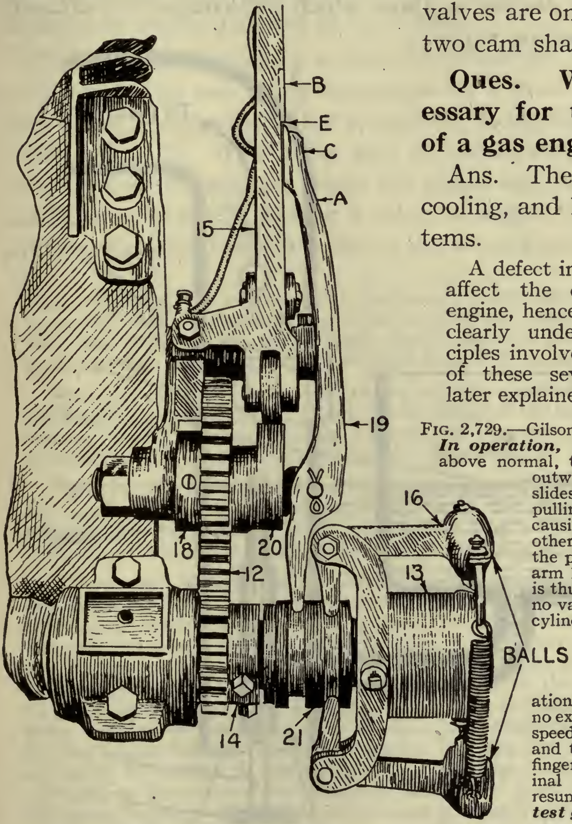


FIG. 2,729.—Gilson hit or miss governor. ***In operation,*** if the speed increase above normal, the governor balls fly outward, and collar 21, slides outward on the shaft, pulling lever 19, with it, and causing the knife on the other end to catch under the projection of the push arm 15. The exhaust valve is thus held open, so that no vacuum can form in the cylinder on the suction stroke. The automatic inlet valve which depends on suction for its operation, does not open, and as no explosion takes place the speed slackens to normal, and the governor balls and finger returns to their original position, the engine resuming operation. ***To test governor,*** turn engine over until the cam 20, has pushed the push rod 15, to its furthest position as shown. Now push the governor lever 19, at A, *in* against the push rod as shown. The knife of the governor lever must pass the projecting plate freely, leaving a clearance of about one-sixteenth of an inch between them at E. If the clearance between these points be insufficient the engine will run unsteady and may possibly stop. To give the proper clearance, remove the governor lever 19, and grind off from the point of knife C, enough to give it the proper clearance.

Carburettors.—A carburettor is a vessel in which the gasoline is vaporized and mixed with air, prior to its introduction into the power cylinder, and the highest fuel efficiency is obtained from the use of such devices as those which will completely saturate the air with gasoline vapor.

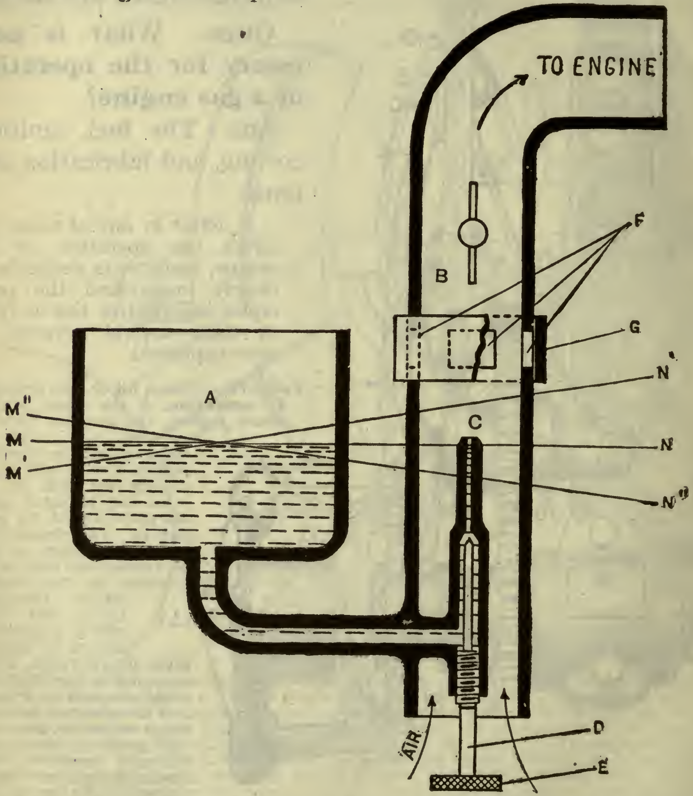


FIG. 2,730.—A rudimentary, or simple form of spray carburetter illustrating the principles of operation employed in the modern device. A, is the receiving chamber; B, the mixing chamber. A connecting passage conveys fuel to the spray nozzle C, controlled by the needle valve D, by turning the thumb wheel E. Air enters through the primary passage in the base and through the auxiliary ports F, the latter being adjustable by the sleeve G, and the mixture to the engine, controlled by a throttle located above the sleeve.

The operation of a carburettor may be clearly explained by considering the simplest form, or what may be called a rudimentary carburettor having a sprayer and means of regulating the mixture by hand as shown in fig. 2,730.

The drawing illustrates a receiving chamber A, and a mixing chamber B, the two being connected by a small passageway or duct which terminates at the sprayer C, made adjustable by the needle valve D. The lower end of the mixing chamber B, is open to the atmosphere, while the upper end is provided with auxiliary air ports F, having a collar or sleeve G, with which to adjust the opening of the ports to the atmosphere.

In explaining the action of this rudimentary carburettor, it is necessary to assume the receiving chamber A, to be filled with gasoline to a level MN, very near the elevation of the spray nozzle, and also to assume the supply replenished as it is used so that the fluid level MN, is maintained.

Now, suppose the upper end of the mixing chamber to be connected with an engine as indicated. Each intake stroke of the engine will displace a volume of air, causing a partial vacuum in the mixing chamber B; the intensity of the vacuum, as will be seen, depending on the engine speed.

Assuming the engine to be working at slow speed with a heavy load and the auxiliary ports F, closed by the sleeve G, the gasoline supply may be adjusted by the needle valve E, so that the engine will receive from the carburettor a mixture of the proper proportion of gasoline vapor and air.*

If now part of the load on the engine be removed so that it will run say twice as fast, the same amount of air and gasoline for each charge must be received by the engine in one-half the time. Under these conditions the mixture would become too rich, that is, too much gasoline would be fed for the amount of air passing through the inlet at the lower end of the mixing chamber. The excess of gasoline is due to the fact that in order to get twice the amount of air through the inlet the suction has to be more than doubled to compensate for the increased frictional resistance set up by the higher velocity of the air passing through the inlet. The suction, or degree of vacuum in the carburettor, being more than doubled, it naturally follows that more than double the amount of gasoline will pass through the spray nozzle. This excess of vacuum is caused by the expansion of the air in entering the carburettor, increasing its velocity *after expansion* more than two fold.

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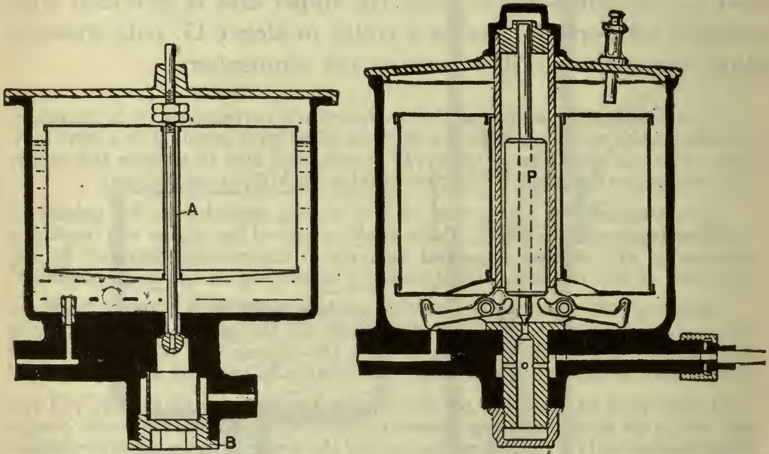
*NOTE.—The fuel charge for a gasoline engine consists of about ten to sixteen parts air to one of gasoline vapor. The proportion varies according to the conditions of the atmosphere, quality of gasoline and conditions of operation.

In order to maintain the mixture in the same proportions for varying demands, it is necessary to provide some means to keep the suction or degree of vacuum in the mixing chamber constant. The suction may be restored to its normal condition by slightly raising the sleeve G, so as to partially open the auxiliary air ports F. This allows some air to enter through the auxiliary ports, thus reducing the velocity of the entering air and relieving somewhat the suction at the lower inlet. The amount of opening of the auxiliary air ports necessary for any change of engine speed may be found by experiment.

By placing a throttle valve in the passage B, between the auxiliary ports and the engine the load may be altered without any variation in the engine speed by adjusting the throttle opening.



FIGS. 2,731 and 2,732.—Two types of float feed. A "float feed" device consists of a cork or hollow metal float placed in the float chamber. It is connected so as to operate the gasoline inlet valve, usually by means of levers. These are arranged in such a manner that, as gasoline enters the float chamber through the inlet valve, the float rises, and in so doing, closes the valve, thus shutting off the supply when the gasoline reaches the desired level. Fig. 2,731 shows a simple direct acting float feed having no levers. As shown, the float is concentric to the inlet valve. Fig. 2,732 shows a type of float feed in which the fuel supply is made adjustable by the use of a variable counterweight P. More modern float feed mechanisms are shown in the cuts following.

In actual construction, automatic devices are employed to maintain the gasoline in the float chamber* at constant level and to adjust the auxiliary port openings to different engine speeds. Carburettors of this type are

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*NOTE.—In discussing the modern carburettor, the term *float chamber* is used instead of receiving chamber as it is generally known by that name, since a float is almost always used to regulate the flow of gasoline into the chamber.

adapted to the requirements of variable speed engines, as auto, and marine engines; for stationary service an auxiliary air valve is not necessary.

The Mixture.—For stationary engines which operate at unvarying speed, the carburettor may supply the mixture of gas and air in constant proportions. With the automobile engine, however, the conditions are quite different, and it has been determined from experience and numerous experiments that a

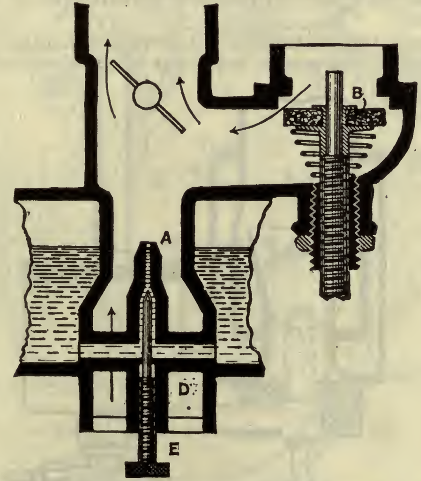

FIG. 2,733.—The ***mixing chamber*** with its appendages, illustrating, in general, the arrangement of parts, the primary and auxiliary air passages; auxiliary valve, spring and adjustment; the spray nozzle with needle valve and the throttle valve. The arrows indicate the direction of the entering air currents and course of mixture. ***In construction,*** the spray nozzle A is located in the center. The adjustable needle valve E regulates the flow of gasoline to the nozzle. The mixing chamber is open to the atmosphere at its lower end D, through which the *primary* or main air supply enters. A *secondary* or *auxiliary* air supply is admitted through the opening to the right, being controlled by the valve B, which is automatic in its action. The lift of this valve may be varied to meet different requirements by the adjustable threaded spindle. ***In operation,*** the pressure in the mixing chamber is lower than that of the atmosphere; the degree of vacuum depends on the amount of throttle opening, the engine speed, and the amount of opening for the primary and secondary air supply. The gasoline, as it is sucked out of the nozzle, made up as it is of hydrocarbons of different values from the point of view of weight and volatility, will hold to the globular form with more or less tenacity, depending upon conditions. That part of the liquid which vaporizes does so through the abstraction of heat from the remainder, which becomes constantly colder. Accordingly, it is desirable that there be provision for supplying heat to the mixture, or preferably to the liquid before it enters the mixing chamber.

constant mixture is not advisable from either the standpoint of fuel economy or best operation. Inasmuch as ignition conditions vary with the speed of the engine and the compression values vary similarly with the throttle opening, it follows that the mixture necessary for maximum power at any given speed differs in accordance with the immediate conditions of combustion.

At low speeds the mixture should be richer than at high. This is due to the fact that at low speeds more heat is lost to the cylinder walls, more compression is lost by leakage, and the combustion can therefore be slower, thus sustaining the pressure. At high speeds the compression is higher, due to less leakage and less loss of heat. Therefore, unless the mixture be leaner at high speed there might be danger of pre-ignition.*

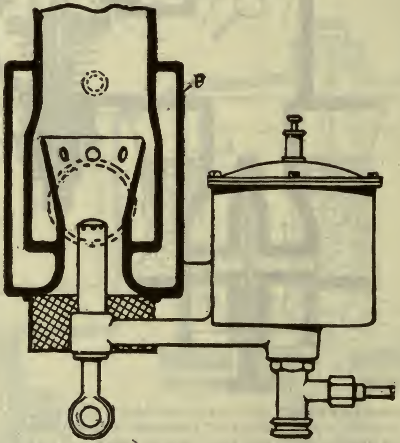

FIG. 2,734.—A jacketed mixing chamber; a ***usual*** but ***objectionable*** method of heating. The mixing chamber as shown, is surrounded by a jacket P, for heating the mixture. This is accomplished either by connection with the cooling water, or exhaust from the engine. During the summer season, when the atmospheric temperature is high, the heating arrangement may be dispensed with. While this method usually improves the engine operation it reduces its power, because with heated *primary air*, the engine cannot take in a full charge. The better way is to heat the gasoline before it enters the mixing chamber, either by, 1, jacketing the float chamber, as in fig. 2,735, or 2, by the use of a superheater as in fig. 2,736.

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*NOTE.—The prefix "pre" before ignition means that the fuel charge is ignited before the time of the spark and has no connection with the stroke or crank position as distinguished from its usage when speaking of steam engines; for example, *pre-admission* means that steam is admitted to the cylinder before the beginning of the stroke.

A lean and highly compressed charge also burns faster and hence gives better pressures and fuel economy than a richer one.

The quantity of mixture that an engine will take varies greatly with the speed.

At slow speeds the volume at approximately carburetter pressure is approximately equal to the cubic content of the cylinders multiplied by the number of power strokes.

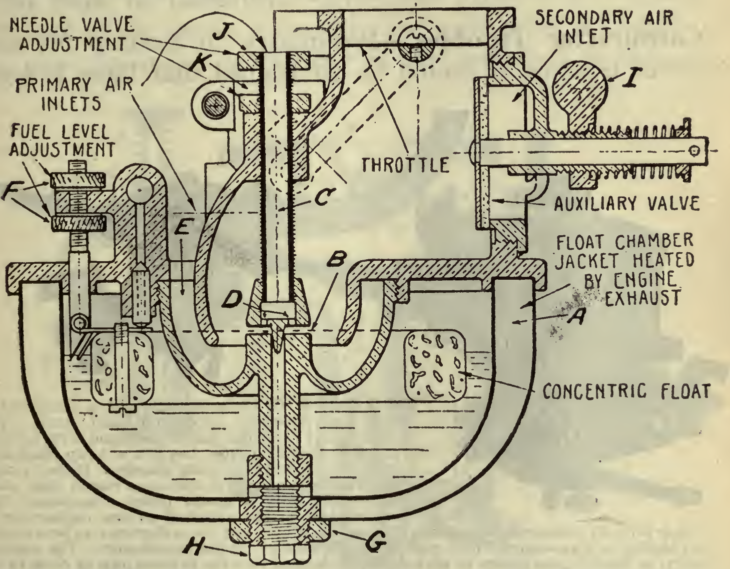


FIG. 2,735.—R. O. C. carburetter with jacketed float chamber. In this carburetter (model D), the *fuel is* ***heated*** *directly in the float chamber before it reaches the spraying nozzle, by passing some exhaust gas through the annular chamber A.* When the fuel is drawn from the nozzle at B, it is met by several fine air jets, passing through the control tube C, and the perforations D, which break the fuel up into a fine spray, which, due to the heat that it contains, immediately flies into a vapor while mixing with the air entering through the annular port E. ***The peculiar construction*** of the port E, gives it the venturi tube principle, the lower half of the tube being practically turned over. The fuel level in this carburetter may be raised or lowered at will, by turning the knurled nuts F, without interfering with the action of the carburetter while the engine is running, and thus the proper level is easily obtained. ***The outer jacket*** can be removed by taking off the nut G, or the float chamber drained by removing the plug H, or the active inner mechanism can be exposed while in position by unscrewing the float chamber, which is merely a drawn brass shell as all connections are part of the cover. The tension of ***the auxiliary valve spring*** is adjusted by turning the wing nut I, without interfering with the action of the valve. ***The needle valve adjustment*** is made by turning the air tube C, by means of its knurled head J, and it is held in any position by tightening the knurled jam nut K. ***An interesting feature*** is that all adjustments, including the removal of the float chamber, are made without even the use of a screw driver. Model A differs from the illustration simply by the absence of the jacket surrounding the float chamber.

At high speeds of one thousand revolutions and over the quantity may drop to less than one-half the theoretical amount, depending on the design of the valves, inlet piping and carburetter passages. This peculiarity reacts upon the compression, and hence on the mixture desired.

The design and class of ignition used have also a marked influence. Poorer mixtures can be used, as the spark is hotter, the throttle can be more nearly closed, resulting in increased engine capacity and fuel economy.

Carburetter Troubles.—Preliminary to hunting for carburetter troubles, it should be ascertained that there is some gasoline in the tank, and that the valve on the pipe leading from same is open.

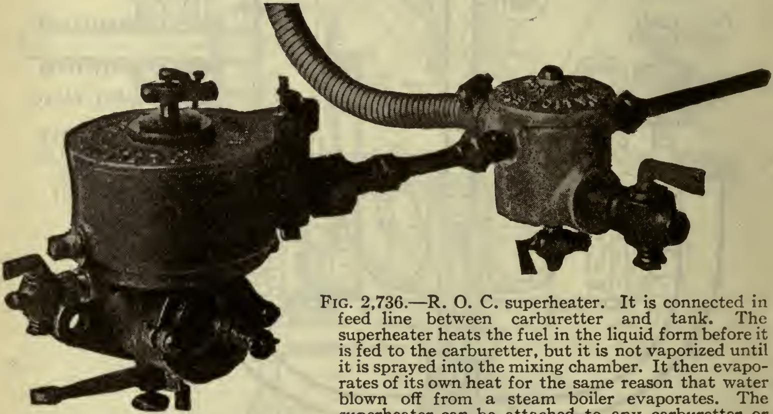


FIG. 2,736.—R. O. C. superheater. It is connected in feed line between carburetter and tank. The superheater heats the fuel in the liquid form before it is fed to the carburetter, but it is not vaporized until it is sprayed into the mixing chamber. It then evaporates of its own heat for the same reason that water blown off from a steam boiler evaporates. The superheater can be attached to any carburetter or mixing valve by connecting it between the gasoline supply and the carburetter, as here shown, and leading a three-eighths inch pipe from the exhaust to the superheater. The exhaust leader, or flexible pipe shown in part should be tapped into the exhaust pipe as close to the engine as possible, and the exhaust outlet from the bottom of the superheater may be run anywhere to the atmosphere.

The carburetter is too often blamed for faulty engine performance, which should be attributed to defects in the ignition system. Such symptoms as fouled plugs, black smoke in the exhaust, etc., point at once to the carburetter, but in cases where such obvious signs are wanting, the ignition system should first be thoroughly examined.

The following carburetter troubles are frequently encountered; they are for the most part due to inattention or carelessness, rather than to defective mechanism.

No Flow of Gasoline.—Sometimes little, if any, gasoline will flow to the nozzle even when the carburetter is flooded in the usual manner. A quantity of dirt sufficient to stop the

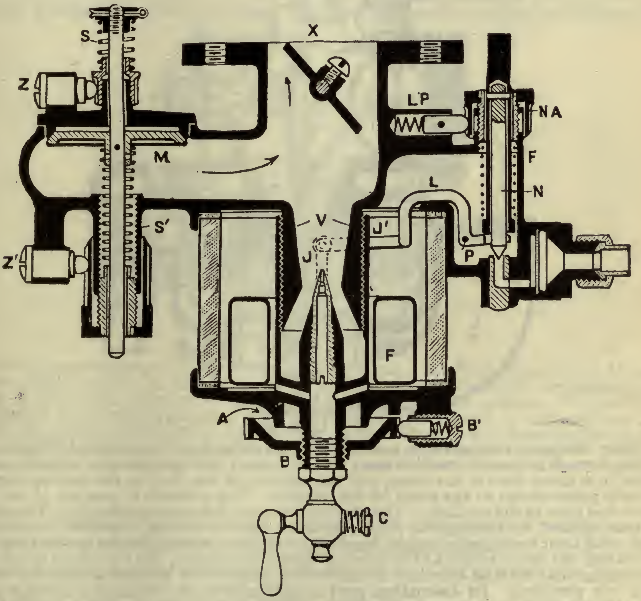


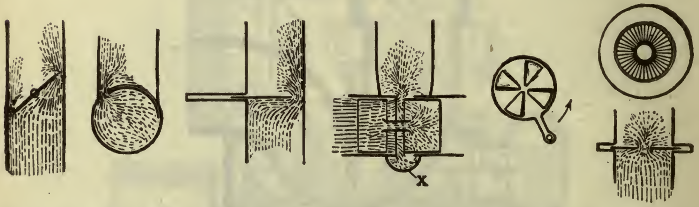
FIG. 2,737.—The Stromberg Carburetter. The principal features of this carburetter are: a glass float chamber, concentric float, venturi shaped mixing chamber, adjustable primary air inlet, and a two spring adjustment for the auxiliary valve. ***In operation*** the gasoline supply is controlled by the float F, through levers J and J′, the latter pivoted at P, and connected to needle valve N. The float is adjustable by the spring I, and nut N A, secured in position by the plunger L P. Primary air enters at A, and is regulated by the adjustable cup B, secured by the plunger B′. The drip cock C, drains the float chamber. The mixture from the venturi tube V, receives supplementary air through the auxiliary valve N, thence it passes to engine through throttle X. The auxiliary valve is controlled by two springs S and S′, the lower one acts on moderate speeds and the upper one on high speeds. The springs have adjusting nuts and self locking devices Z and Z′.

located by heating the float and passing a lighted match over the surface, which will ignite the issuing vapor.

To repair, the hole is enlarged with an awl, the float drained, and soldered.

Leaking Tank.—Tanks are liable to leak through the opening of the seams by jarring or vibration.

Galvanized iron tanks, such as are furnished on some machines, should be discarded when a leak results from rust, as it is practically no use to solder it. A heavy gauge copper tank should be substituted.



FIGS. 2,740 to 2,746.—Several designs of throttle valves are here shown partially opened, where the effects of separation and deflection of the liquid globules are illustrated for each case. It will be seen that all these throttles act as separators when not fully open. The first four throw the liquid upon the walls unevenly. The last one shown to the right, while better distributing the liquid over the walls, is like the rest, an energetic separator. Throttles like the one shown in fig. 2, 743 are rarely found in later designs. Probably all who have had any experience with this type remember that a drain cock is inserted at X.

The supply pipe should be made flexible by a loop to avoid strains due to vibration. All soldered connections should be inspected from time to time.

Loss of Buoyancy.—A cork float sometimes loses its buoyancy by becoming saturated with gasoline.

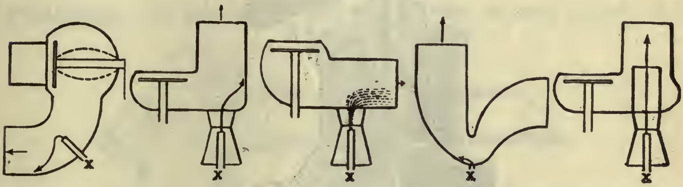
It should be removed and thoroughly dried by placing the float in a warm place; after drying, a coat of shellac should be applied.

Impure Gasoline.—Many carburetter troubles would be avoided if more care were taken to free gasoline of all dirt before its entrance into the tank.

When filling the tank, a strainer funnel should be used. A piece of chamois skin makes an excellent filter; if a wire gauze be used it should have a very fine mesh. In the absence of a strainer funnel, three or four layers of fine linen fitted inside an ordinary funnel may be used. The same funnel should never be used for both gasoline and water.

Stale Gasoline.—When a car is not used for some time the gasoline in the float chamber loses its strength.

If the engine should not start, the tank valve should be closed, and the carburetter drained through the pet cock, which is usually provided in the bottom of the chamber for this purpose. When empty, the pet cock should be closed and the tank valve opened, not forgetting to give the float chamber time to fill before trying to start the engine.



FIGS. 2,747 to 2,751.—A few examples of carburetter design, showing courses followed by the fuel after leaving the nozzle, for different arrangements of parts and passages. The heavy arrowed line leading from the nozzle X indicates the course taken by the mixture as influenced by the design. These diagrams are self explanatory, and show that liquid globules are precipitated against some portions of the mixing chamber wall almost immediately after leaving the nozzle. This tends to disturb the homogeneity of the mixture, and requires that provision be made for correcting this effect in the remaining portions of the passage.

Low Grade Gasoline.—This sometimes causes the engine to misfire and not develop its full power. Inferior fuel is generally indicated by a smoky exhaust and a disagreeable odor. Gasoline suitable for automobile use should test 76 degrees, or not much below this gravity, although the kind now furnished or so called gasoline, is almost the same as kerosene.

Water in Gasoline.—This is generally indicated when the engine runs irregularly, and finally stops.

To test, a small quantity of the gasoline is placed on a clean knife blade or other smooth metallic surface. The gasoline will evaporate, and if water be present it will collect in small globules unless the water has been purposely ***chemically*** combined with the gasoline. Gasoline and water chemically combined will burn slowly with a yellowish flame.

Freezing of Carburetter.—When water enters the float chamber it settles to the bottom and in cold weather prevents the action of the float by freezing; the water is also liable to enter the spray nozzle where it may congeal. When heavy demands are made on a carburetter it becomes very cold, as the heat required to effect evaporation is much more than that

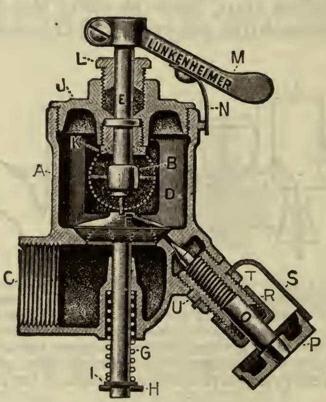


FIG. 2,752.—A vaporizer or generator valve. This differs from a carburetter in the absence of a float chamber, and consists of a mixing chamber containing a check valve and having: 1, an air inlet; 2, a gasoline inlet; and 3, an exit to engine. ***Its operation,*** is as follows: On the suction stroke, the partial vacuum produced in the mixing chamber A, permits the atmospheric pressure to act upon the valve F, opening same against the tension of spring G, which is held in position by the washer I, and cotter H. During this period, the gasoline valve in the port seat is uncovered and a small amount of gasoline is sprayed into the incoming volume of air and passes into the mixing chamber where the mixing is further assisted, in some designs, by baffle walls. At the end of the suction stroke the pressure in the mixing chamber becomes equalized with the atmosphere and the spring causes the valve F, to seat, thereby retaining the mixture and shutting off any further injection of gasoline or air. The gasoline supply may be adjusted by the needle valve O, operated by the thumb wheel P, which has a flat spot on its cirqumference on which the spring S, bears to retain the adjustment. The spring can be turned to any position by loosening the lock nut T. The volume of mixture to the engine is regulated by a sliding throttle D, operated by lever M, and locked by spring N, which engages notches in a graduated dial. A vaporizer when used on a two cycle engine requires no check valve between it and the engine.

available from the entering air. Under extreme conditions moisture is deposited in the form of frost, indicating a temperature in the carburetter too low for good working.

These conditions may be avoided by jacketing or heating the air supply, or preferably by pre-heating the gasoline.

Cold Weather.—In extremely cold weather it may be necessary to warm the carburetter and admission pipe. This may be done by pouring boiling water over them.

Cranking.—So far as carburetter action is concerned, ***a few quick turns of the crank*** will be more likely to start the engine than ten minutes or more of slow grinding.

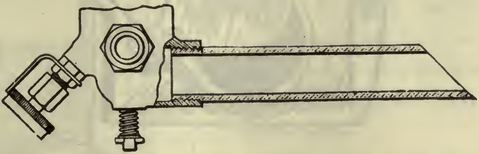


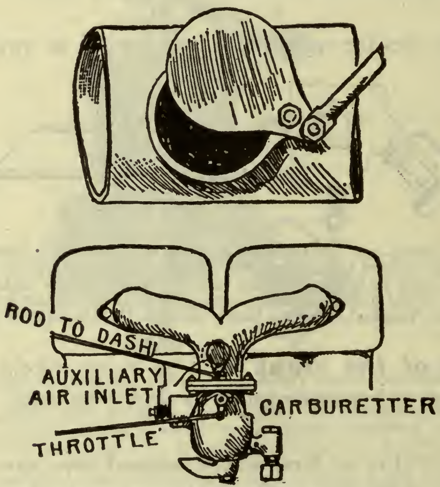
FIG. 2,753.—An air inlet pipe. This consists of a short length of pipe threaded at one end and screwed into the air inlet of a vaporizer. A saving in fuel is secured by its use as any gasoline or vapor that may be blown into the inlet when the valve seats, is retained in the pipe and drawn into the mixing chamber during the next suction stroke. Without any extension of the inlet, this fuel would be blown out into the atmosphere and lost.

Misfiring.—This is frequently caused by too weak, or too rich a mixture. Misfiring allows the unburnt charge to accumulate in the exhaust pipe and muffler; sometimes accumulated gas is ignited by a later charge, causing a very loud report like a tire explosion. Misfiring on slow speed may be caused by too weak a mixture due to having the float set too low, or by leaks in the pipe and connections between the throttle and the engine.

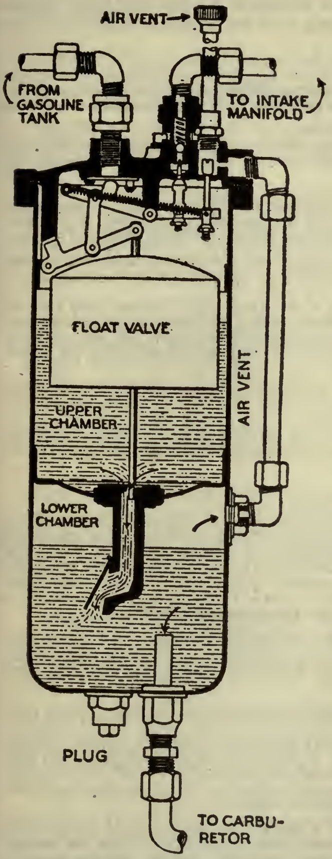
After Firing.—This is usually caused by the delayed ignition or combustion of the previous charge, due to a mixture that is too rich or too weak, hence it burns slowly with continued combustion after passing into the exhaust.

Weak Explosions.—Quite regular, but weak explosions ma be due to either too rich, or too poor a mixture, or to the loss o compression.

A hiss inside the cylinder indicates a leaky piston ring, or tha the openings of the piston rings are in line.



Figs. 2,754 and 2,755.—Hand control supplementary air supply for automobile service. Th simple device shown can and should be applied to any car with little trouble. Cut an opening at any suitable place in the pipe between the carburetter and engine and cover it with a sliding collar, valve, flap, or any other device that can be easily opened or closed by a leve on the steering column, the essential feature being that it must be fairly well air tight in th closed position. With a little experience on the road, the driver will soon discover th point of engine speed determined by the throttle lever at which he can commence to ope the extra air supply. If he has never before driven a car so fitted it will be a revelation t him, for this extra air port can be opened wider and wider to an astonishing extent befor the engine will misfire to indicate that the mixture is too weak. This extra air port beside serving as a power increaser, can be made to act as a scavenger and cylinder cooling agent When descending a long hill, by switching off the spark, entirely closing the throttle an opening the extra air port to its full extent (the top speed gear and clutch are of course kep in engagement so that the car is driving the engine), cold air is drawn into the engine o each suction stroke, clearing out every particle of hot gases and helping materially not onl to cool the engine and spark plugs but also to keep the points of the latter much cleane and freer from carbonized oil than would otherwise be the case.



A little soapy water around the relief cock, spark plug, or other opening into the combustion space will indicate a loss of compression by the formation of bubbles.

Ignition.—Most treatises on ignition begin with an explanation of electrical principles and considerable space is thus taken up, which, if confined to the main subject, would be of greater value to the reader, assuming that he either has an elementary knowledge of electricity, or that he will acquire this knowledge elsewhere.

For convenience of the reader a very condensed summary of the electrical principles employed in ignition is here given.

Fig. 2,756.—Stewart vacuum gasoline system, for use where the gasoline tank is lower than the carburetter. The vacuum tank is divided into two chambers, upper and lower. The upper chamber is connected to the inlet manifold while another pipe connects it with the main gasoline supply tank. The lower chamber is connected to the carburetter. The intake strokes of the engine create a vacuum in the upper chamber of the tank, and this vacuum draws gasoline from the supply tank. As the gasoline flows into this upper chamber it raises a float valve. When this float valve reaches a certain height it automatically shuts off the vacuum valve, and opens an atmospheric valve, which lets the gasoline flow down into the lower chamber. The float valve in the upper chamber drops with the gasoline flowing out, and when it reaches a certain point it in turn reopens the vacuum valve, and the process of refilling the upper chamber begins again. The same processes are repeated continuously and absolutely automatically. The lower chamber is always open to the atmosphere, so that gasoline flows to the carburetter, as required, uninterruptedly and at an even pressure.

Electricity.—The name electricity is applied to an invisible agent known only by the effects which it produces, and the various ways in which it manifests itself.

Electrical *currents* are said to flow through *conductors*. These offer more or less *resistance* to the flow, depending on the material. Copper wire is generally used as it offers little resistance to the flow of the current.

The current must have pressure to overcome the resistance of the conductor and flow. This pressure is called *voltage* caused by what is known as *difference of pressure* between the source and terminal.

An electric current has often been compared to water flowing through a pipe. The pressure under which the current flows is measured in *volts* and the quantity that passes in *amperes*. The resistance with which the current meets in flowing along the conductor is measured in *ohms*.

The flow of the current is proportional to the voltage and inversely proportional to the resistance. The latter depends upon the material, length and diameter of the conductor.

Since the current will always flow along the path of least resistance it must be so guarded that there will be no leakage. Hence to prevent leakage, wires are *insulated*, that is, covered by wrapping them with cotton or silk thread or other insulating materials. If the insulation be not effective, the current may leak, and so return to the source without doing its worth. This is known as a *short circuit*.

The conductor which receives the current from the source is called the *lead* and the one by which it flows back, the *return*.

When wires are used for both lead and return, it is called a *metallic circuit;* when the metal of the engine is used for the return, it is called a *grounded circuit*, the term originating in telegraphy, where the earth is used for the return.

In ignition diagrams, then, the expression "to ground" means *to the metal of the engine.*

An electric current may do work of various kinds, but the one property which makes it available for ignition is the fact that whenever its motion is stopped by interposing a resistance, the energy of its flow is converted into heat. In practice this is accomplished in two ways: 1, by suddenly breaking a circuit; 2, by placing in the circuit a permanent *air gap* which the current must jump. In either case, the intense heat caused by the enormous resistance interposed, produces a spark which is utilized to ignite the charge. The first method is known as the ***make and break*** or *low tension* and the second, the ***jump spark*** or *high tension.*

An electric current is said to be: 1, *direct*, when it is of unvarying direction; 2, *alternating*, when it flows rapidly to and fro in opposite directions; 3, *primary*, when it comes directly from the source; 4, *secondary*, when the voltage and amperage of a primary current have been changed by an *induction coil.*

A current is spoken of as *low tension*, or *high tension* according as the voltage is low or high.

A high tension current *is capable of forcing its way against considerable resistance, whereas* **a low tension current** *must have its path made easy.* A continuous metal path is an easy one, but an interruption in the metal, as, the permanent air gap of a spark plug, is difficult to *bridge*, because air is a very poor conductor. Air is such a poor conductor that it is usually, though erroneously, spoken of as a *non-conductor;* it is properly an insulator.

The low tension current is only able to produce a spark when parts are provided in the path, so arranged that they may be in contact and then suddenly separated. The low tension current will, as the separation occurs, tear off very small metallic particles and use these as a *bridge* to keep the path complete. Such a bridge is called *an arc*, the heat of which is used for ignition.

Magnetism.—The ancients applied the word "magnet," *magnes lapes*, to certain hard black stones which possess the property of attracting small pieces of iron, and as discovered later, to have the still more remarkable property of pointing north and south when hung up by a string; at this time the magnet received the name *lodestone.*

The automobile word magneto is derived, as may easily be understood, from the word *magnet.*

Magnets have two opposite kinds of magnetism or magnetic poles, which attract or repel each other in much the same way as would two opposite kinds of electrification.

One of these kinds of magnetism has a tendency to move toward the north and the other, toward the south.

The two regions, in which the magnetic property is strongest, are called the *poles.* In a long shaped magnet it resides in the ends, while all around the magnet half way between the poles there is no attraction at all. The poles of a magnet are usually spoken of as *north pole* and *south pole.*

When a current of electricity passes through a wire, a certain change is produced in the surrounding space producing what is known as a *magnetic field.*

If the wire be insulated with a covering and coiled around a soft iron rod, it becomes an electromagnet having a north and south pole, *so long as the current continues to flow.* The magnetic strength increases with the number of turns of the coil, for each turn adds its magnetic field to that of the other turns.

Induction.—If a second coil of wire be wound around the coil of an electromagnet, but not touching it, an *induced current* is produced in this second coil by what is known as *induction*, each time the current in the inside coil begins or ceases flowing. The inside coil is called the *primary winding* and the outside coil the *secondary winding.* Similarly, the current passing through the inside coil is called the *primary current* and that in the outside coil the *secondary* or *induced current.*

It has been found that by varying the ratio of the number of turns in the two coils, the ratio of voltage of the two currents is changed approximately proportionately. That is, if the primary winding be composed of ten turns and the secondary, of one hundred, the voltage of the secondary current is increased approximately ten times that of the primary. This principle is employed to produce the extremely high tension current necessary with the jump spark method of ignition.

Methods of Producing Electricity.—Currents are produced by, 1, chemical, and 2, mechanical means. In the first method, two dissimilar metals such as copper and zinc called *electrodes* are immersed in an exciting fluid or *dielectric.* When the electrodes are connected at their terminals by a wire or conductor, a chemical action takes place, producing a current which flows in the external circuit from the copper to the zinc. This device is called a *cell*, and the combination of two or more of them connected so as to form a unit, is known as a *battery.*

The word battery is frequently used *incorrectly* for a single cell. That terminal of the copper electrode from which the current flows is called a *plus* or *positive pole* and the zinc electrode terminal a *negative pole.* It should be carefully noted, however, that the copper electrode itself is negative and the zinc electrode, positive.

Cells are said to be *primary* or *secondary* according as they generate a current of themselves or first require to be charged from an external source, storing up a current supply which is afterwards yielded in the reverse direction to that of the charging current.

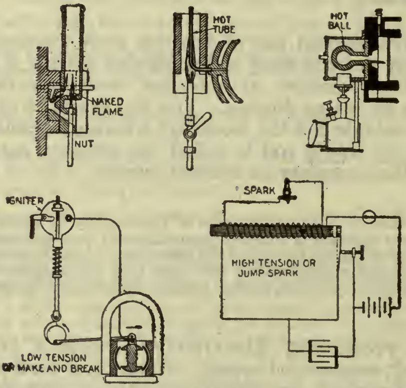
There are two methods of producing an electric current by mechanical means, 1, by a *dynamo*, and 2, by a *magneto.* A dynamo has an *electromagnet* which is known as a *field magnet* to produce a *magnetic field* and an armature which when revolved in the magnetic field develops electric current. A magneto has a permanent magnet to produce the magnetic field and an armature which is usually arranged to revolve between the poles of the magnet.

The basic principles upon which dynamos and magnetos operate are the same.

Magnetos are divided into two classes, 1, *low tension*, and 2, *high tension* according as they generate a current of low or high voltage. Low tension magnetos are used for make and break ignition and the high tension type for the jump spark system. There are numerous so called high tension magnetos on the market each consisting of a low tension magneto in combination with a secondary induction coil used to produce a high tension spark.

Methods of Ignition.—The charge in the cylinder of a gas engine may be ignited in several ways, as

1. By means of a naked flame;
2. By means of a highly heated metallic surface;



FIGS. 2,757 to 2,761.—Various methods of ignition. Fig. 2,757, naked flame; fig. 2,758, hot tube; fig. 2,759, hot ball; fig. 2,760, low tension electric or make and break; fig. 2,761, high tension electric or jump spark.

3. By an electric spark;
4. By the heat of a very high compression.

The naked flame is practically obsolete, and the hot surface or hot tube is used to a very limited extent, except in the case of some types of oil engine. Many builders of standard engines, however, are prepared to furnish hot tube ignition, though electric ignition represents the standard practice.

There are two electric systems in general use for igniting the charge by electricity.

1. The low tension, or *make and break;*
2. The high tension, or *jump spark.*

The low tension system is electrically simple and mechanically complex, while the high tension system is electrically complicated and mechanically simple.

Low Tension Ignition.—In this system there is a device known as an *igniter*, placed in the combustion space of the engine cylinder. This consists of two electrodes, or spark terminals, one of which is stationary and the other movable. The stationary electrode is insulated, while the other having an arm within the cylinder and placed conveniently near is capable of being moved from the outside so that the arm comes into contact with the stationary electrode and separates from the latter with great rapidity. This sudden breaking of the circuit produces an electric arc or *primary spark* caused by the inductance, that

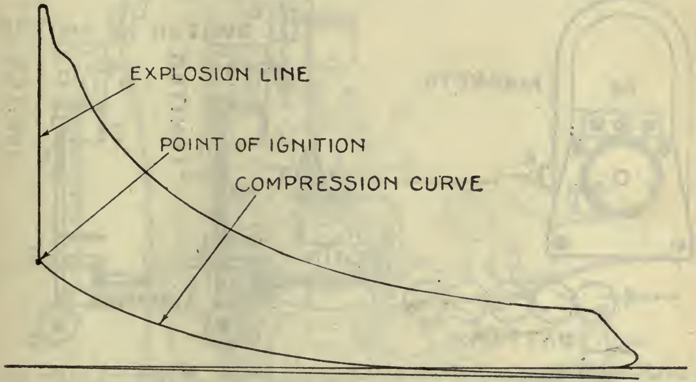


Fig. 2,762.—Indicator card for gas engine illustrating the ***"point of ignition."*** It will be noted that compression continued to the end of the stroke, before the compression curve made an abrupt change into a nearly vertical line, the point of ignition, that is, the piston position at the instant of the spark, the nearly vertical "explosion" line with the high peak coming almost to a point, denotes a strong mixture and a quick explosion.

is, by the "inertia" or tendency of the current to continue flowing after the separation of the contact points.

The current may be derived from either a primary battery, storage battery or low tension magneto.

While it is possible to produce a spark by simply breaking a battery circuit, it is necessary in order to have a spark of sufficient intensity and duration to introduce into the circuit a *primary induction coil* consisting of a long iron core wound with a considerable length of low resistance insulated copper wire. When a magneto is used, the coil is not necessary as the armature winding serves the same purpose. A magneto furnishing either direct or alternating current may be used; the voltage will depend on the armature speed and the strength of the magnets.

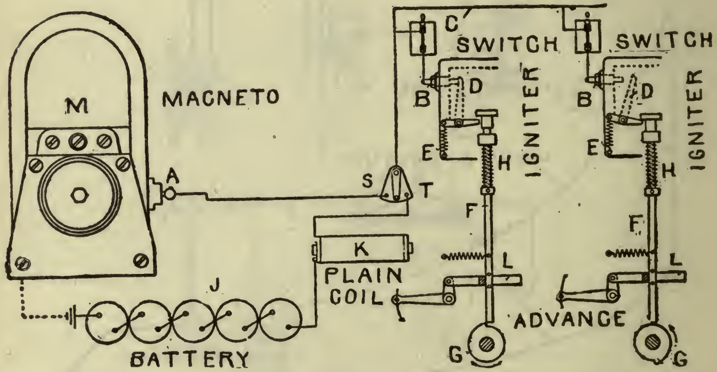


FIG. 2,763.—A low tension or make and break ignition system. In operation, as the nose of the cam G, passes rod F, the latter suddenly drops by the action of spring H. The head of the rod, which has been raised by the cam somewhat above the arm of D, will in its descent strike D, a blow which abruptly breaks contact between D and B, thus producing a spark. When not acted upon by the head of the rod F, D, is held in contact with B, by the Spring E. The system is explained in detail in the text.

Iridium or platinum is used for contact points of the electrodes, as these metals resist the oxidizing effect of electricity and heat better than others.

In low tension ignition a considerable interval of time is required for the current to rise to its full value and the time of separation of the electrodes should not be sooner than the moment when the maximum current strength has been attained. When a magneto is used the current strength increases with the speed, hence the contact interval can be shorter at high speeds than when a battery is used.

The operation of a low tension ignition system is illustrated in fig. 2,763.

At the instant the circuit is broken by the separation of the contact points, the self-induction thus set up in the coil K, opposes any rapid change in the current strength, hence, the current continues to flow momentarily after the circuit is broken, resulting in a *primary spark*. The action is the same as though the current possessed the property of "inertia," that is, time and resistance both are necessary to bring it to a state of rest. This inertia effect is intensified by the action of the induction coil. When a magneto is used, the armature windings serve the same purpose.

The timing of the spark is accomplished by the adjustable guides L, which

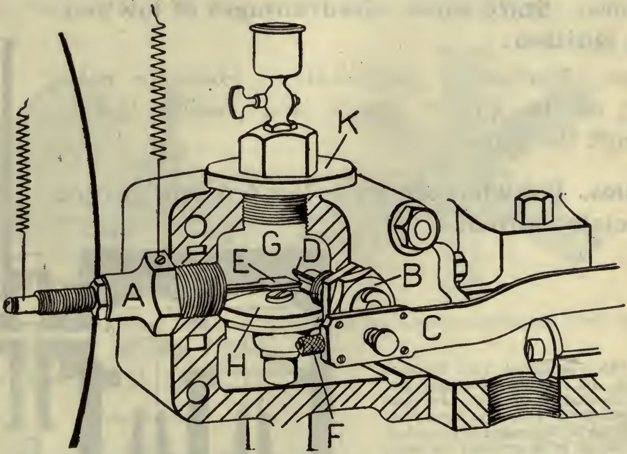


FIG. 2,764.—***Wipe contact*** igniter. It consists of two independent electrodes, the stationary electrode A, and the movable electrode B. When the latter is revolved by the motion of the igniter rod C, the revolving blade D, is brought into contact with the spring E, at each rotation and produces the spark. In this arrangement the break is more effective than in the hammer break type, and gives a larger spark with a given battery capacity, while at the same time, the wiping contact of the two parts prevents the accumulation of burnt carbon or scale on their edges, and thus serves to keep the contact surfaces bright and clean. On the other hand, it possesses the drawbacks incident to the use of a spring which is exposed to the extremely high temperatures developed within the cylinder. The use of flat springs is particularly disadvantageous as it is very difficult to temper them uniformly, and they are consequently liable to break without warning. Furthermore, the wear of the electrodes is excessive. The moment of ignition can be adjusted while the engine is running by turning the thumb screw F, on the end of the igniter rod, and this screw is used also to retard the impulses at starting, thus preventing the engine moving backwards. The igniter is located in the inlet chamber G, directly over the head of the admission valve H, and either one of the electrodes can be reached for inspection or removal, independently, by simply removing the cap K.

serve to vary the horizontal position of the lower ends of the rods F, and thus vary the instant at which their ends pass the nose of each cam.

In make and break ignition it is necessary in order to produce a good spark, that the "break" or separation of the contact points of the igniter should take place with **extreme rapidity,** that is the spring H (fig. 2,763), should be sufficiently strong to cause the shoulder or rod F, when it falls, to strike the igniter arm a decided blow, thus quickly snapping apart the contact points.

Ques. State some disadvantages of low tension ignition.

Ans. Mechanical complication, excessive noise, wear of the igniter points, and possible leakage through the igniter.

Ques. For what service is low tension ignition especially suited?

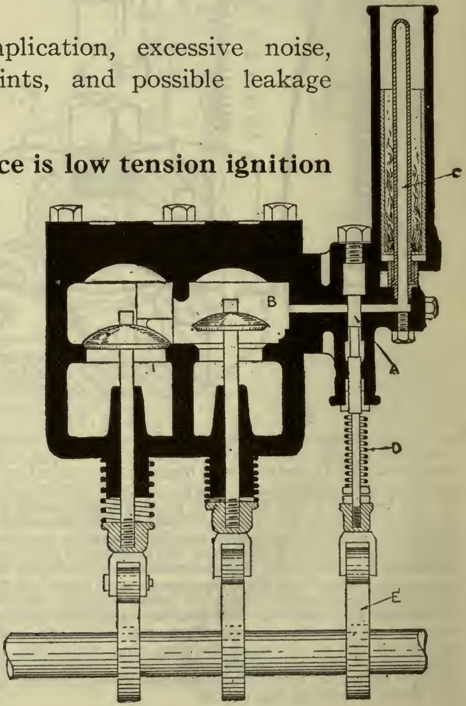


FIG. 2,765.—Sectional view through valves of engine showing hot tube method of ignition. ***In construction,*** a valve A, commonly called the timing valve, is provided, and which is interposed between the admission valve chamber B, (communicating with the clearance space of the cylinder), and the interior of the hot tube C. This valve is normally held closed by the spring D. When the piston reaches its inner dead point at the end of the compression stroke, a cam E, on the secondary shaft, opens the valve and allows a portion of the compressed charge to pass into the hot tube where it ignites. The timing valve is held open throughout the power and exhaust strokes, thus permitting the products of combustion to be carried out of the tube with the exhaust.

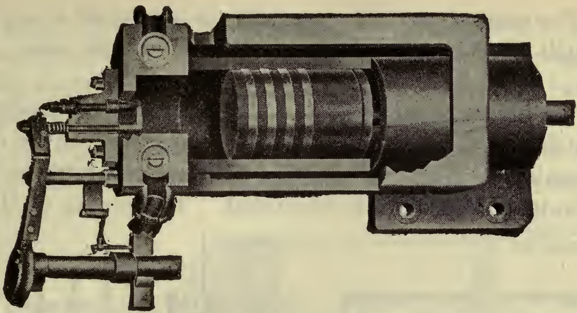


FIG. 2,766.—Diagram of make and break ignition system as applied to the Domestic engine.

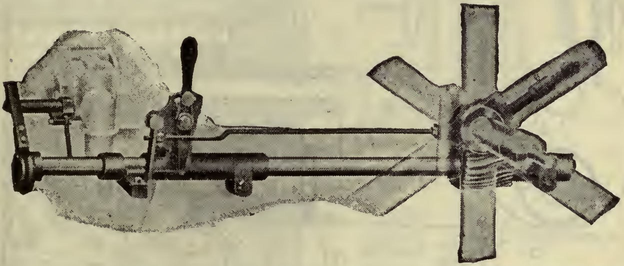


FIG. 2,767.—Sectional view of Domestic cylinder and head showing make and break ignition mechanism.

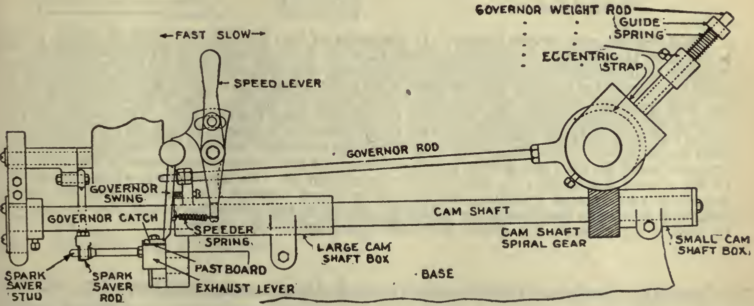


FIG. 2,768.—Domestic igniter arm, governor rod and speed lever.

Ans. For marine service especially in open, off shore fishing boats, such as Cape Cod dories, Sea Bright skiffs, etc.

High Tension Ignition.—In this method of producing a spark, a device called a *spark plug* is employed. This consists of two stationary electrodes, one of which is grounded to the engine cylinder and the other insulated. The points of the electrodes are permanently separated from each other by about ¹⁄₃₂ of an inch, the space between the points being known as

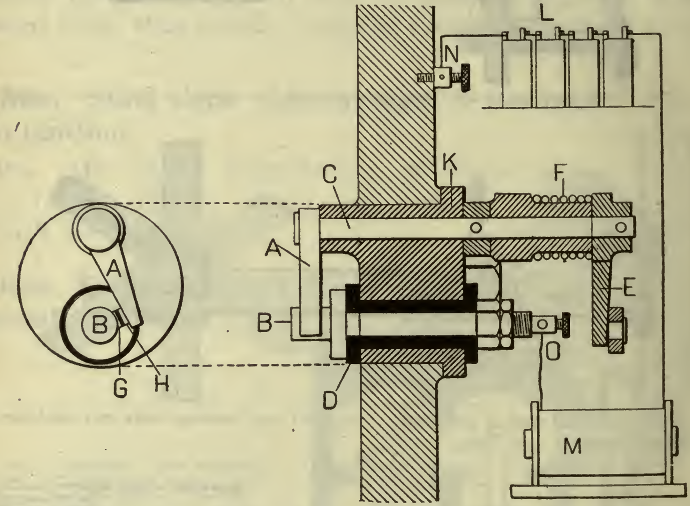


FIG. 2,769.—***Hammer break*** igniter. ***It consists of*** two metallic terminals A and B. The terminal A, is mounted on a movable shaft C, while B, is stationary and ***insulated*** from the cylinder wall by the lava bushing D. A suitable cam rod, attached to the crank E, provides the means for rocking the terminal, A, so as to bring it in contact with the terminal B, *and then **quickly** separate* the terminals to produce the spark. The helical spring F, provides a semi-flexible connection between the shaft C, and the crank E. The contact points of the two terminals are tipped with two small pieces of platinum G and H, and both terminals are mounted in the removable plug K, which is usually inserted through the wall of the cylinder head, so that the igniter points extend into the compression space of the cylinder. In the circuit is a battery L, and primary spark coil M. ***In operation,*** when the igniter terminals are brought together, the circuit is closed through the battery and the spark coil, and when the terminals are ***quickly separated***, the self-induction of the coil causes an electric arc between the igniter terminals which ignites the charge.

an *air gap*. This space offers so much resistance to the flow of an electric current that a very high pressure is required to cause the current to burst through the air gap and produce a spark, hence the term "high tension ignition."

Since the spark jumps from one electrode to the other, this method of igniting the charge is also known as the *jump spark* system. The spark itself is properly described by the prefix *high tension* or *secondary*.

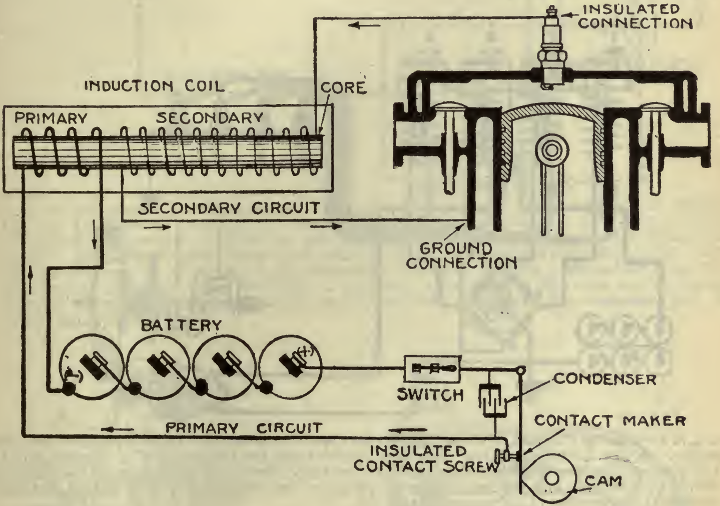


FIG. 2,770.—Diagram illustrating the principles of high tension or jump spark ignition. The nose of the cam in revolving engages the contact maker which completes the primary circuit and allows current to flow from the battery through the primary winding of the coil; this magnetizes the core. The primary circuit is now broken by the action of the cam and magnetic changes take place in the coil which induce a momentary high tension current in the secondary circuit. The great pressure of this current forces it across the air gap of the spark plug and as it bridges the gap a spark is produced. The arrows indicate the paths of the currents.

In the production of the spark two distinct circuits are necessary: 1, a low tension or *primary* circuit, and 2, a high tension or *secondary* circuit. The current which flows through the low tension circuit is called the *primary* *current*, and that which it *induces* in the high tension circuit, the *secondary current.*

In order to obtain the high pressure required to produce a spark, a device known as a *secondary induction coil* is used which transforms the primary current of low voltage and high amperage into a secondary current of high voltage and low amperage, that is, the quantity of the current is decreased and its pressure increased.

The general principles upon which high tension or jump spark ignition is based are as follows:

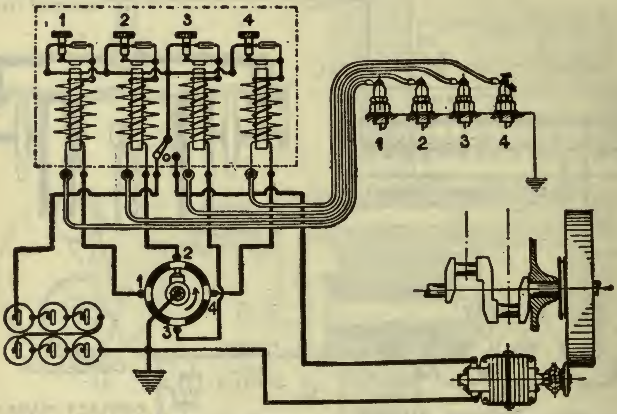


FIG. 2,771.—Wiring diagram of a dual jump spark ignition system for a four cylinder four cycle engine. A dry battery and low tension magneto form the two sources of current supply. The primary, or low tension circuit is shown by heavy lines, the secondary or high tension circuit by fine lines and the leads to spark plugs by the double lines. The dotted rectangle represents the outline of a four unit dash coil.

An automatic device is placed in the primary circuit which closes and opens it at the time a spark is required. When the circuit is closed, the primary current flows through the primary winding of the coil and causes a secondary current to be induced in the secondary winding. The spark plug being included in the secondary circuit opposes the flow of the current by the high resistance of its air gap. Since the pressure of the secondary current is sufficient to overcome this resistance, it flows or "jumps" across the gap and in so doing intense heat is produced resulting in a spark.

Sometimes the spark is obtained by keeping the primary circuit closed except during the brief interval necessary for the passage of the spark at the plug points. A secondary spark, then, may be produced by either open or closed circuit working, that is, the primary circuit may be kept either opened or closed during the intervals between sparks.

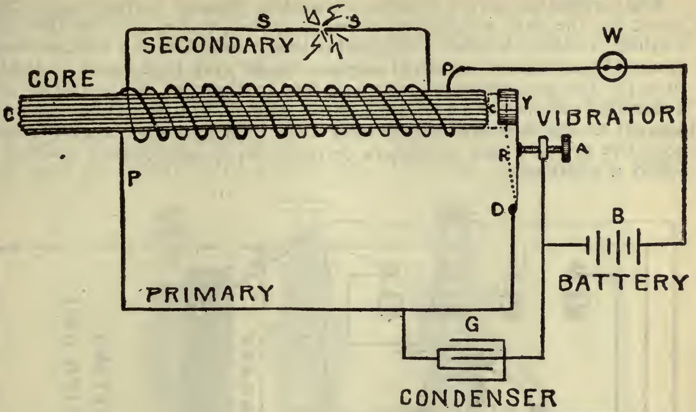
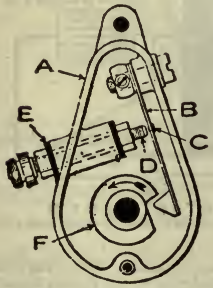


FIG. 2,772.—Diagram of a vibrator coil. The parts are as follows: A, screw contact; B, battery; C, core; D, vibrator terminal; G, condenser; P, primary winding; S, secondary winding; W, switch; Y, vibrator. When the switch is closed, the tollowing cycle of actions takes place: 1, the primary current flows and magetizes core, 2, magnetized core attracts the vibrator and breaks primary circuit, 3, the magnetism vanishes, inducing a momentary high tension current in the secondary winding, 4, magnetic attraction of the core having ceased, vibrator spring re-establishes contact, and 5, primary circuit is again completed and the cycle begins anew.



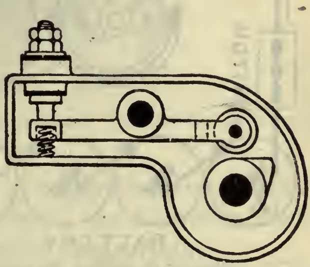


FIG. 2,773.—A contact maker and mechanical vibrator or ***trembler***. The case A, is usually attached to the gear box of the engine; B, is the blade; C, a platinum contact point; D, an insulated adjusting screw; E, a bushing with insulation; F, the operating cam. As this cam revolves, the weight on the end of blade B, drops into the recess on the cam, causing the blade to vibrate and make a number of contacts with D, thus producing a series of sparks when in operation.

FIG. 2,774.—A contact breaker. This device keeps the circuit closed at all times except during the brief interval necessary for passages of the spark at the plug points. Used to advantage on engines running at very high speeds, as it allows time for the magnetic flux in the core of the coil to attain a density sufficient to produce a good spark.

The automatic device which controls the primary current to produce a spark by the first method is called a *contact maker*, and by the second method, a *contact breaker*. A closed primary circuit with a contact breaker is used to advantage on small engines run at very high speed as it allows time for the magnetism or magnetic flux in the core of the coil to attain a density sufficient to produce a good spark. The word *timer* is usually applied to any device which controls the primary current, when it controls both the primary and secondary currents, as in *synchronous ignition* it is called a *distributer*.

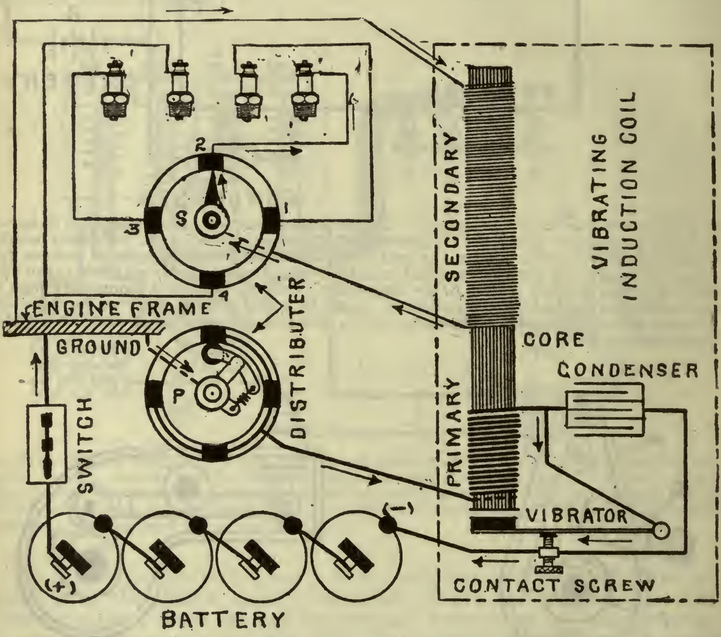
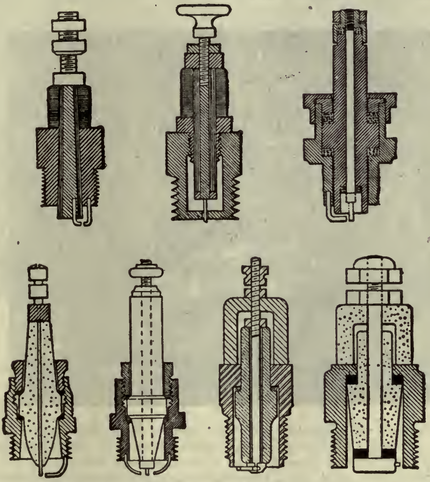


FIG. 2,775.—Diagram illustrating the principles of synchronous ignition. **For clearness** the primary and secondary elements of both the coil and the distributer are shown separated. When the primary rotor of the distributer completes the primary circuit, current from the battery flows and the vibrator operates, making and breaking the current with great frequency. A high tension current, made up of a series of impulses, is induced in the secondary circuit and distributed by the rotor arm during its revolution to the several cylinders in the proper order of firing.

Ignition Troubles.—To successfully cope with ignition troubles there are two requisites, 1, a thorough knowledge of the system used and 2, a well ordered course of procedure in looking for the source of trouble. In many ignition systems, the chief difficulty encountered in the location of defects arises from the fact that faults in different portions of the circuit sometimes make themselves manifest by the same symptoms. If each defect had its individual symptom, locating the trouble would be comparatively easy, but as it is, it is sometimes quite difficult to find the defective parts. In general the following method should be adopted to locate possible derangements:



FIGS. 2,776 to 2,782.—Sections of well known spark plugs. The first five have porcelain insulation; the last two, mica.

1. Examine the source of current supply; if a battery, test each cell separately and remove any found to be weak. When a magneto is used, disconnect the drive and turn armature by hand, if the field magnets have not lost their strength the armature should turn perceptibly hard during certain portions of each revolution.

2. Examine the primary circuit for breaks in its continuity, see that all connections are bright and firmly held together by the binding screws; the timer contacts should be clean.

3. The spark plug points should be clean and the air gap the proper length—about one thirty-second of an inch, or the thickness of a dime.

4. See that the vibrator contacts are in good condition and the adjustment correct. With this preliminary examination the system may now be tested.

Testing the Spark Plug.—The spark plug should be unscrewed and placed on the cylinder without disconnecting the wire to the insulated electrode; the body of the plug only should touch the metal of the cylinder.

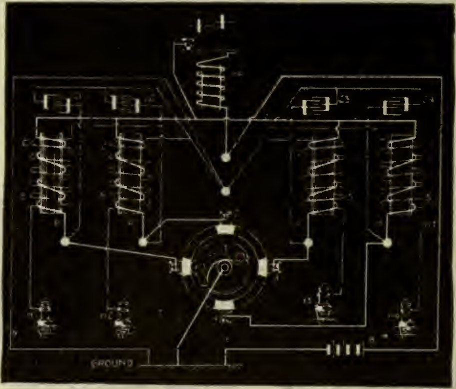


FIG. 2,783.—Circuit diagram of a master vibrator coil. B, is the battery; C, the unit coils; C1, C2, etc., the condensers; P, the primary windings and S, the secondary windings; H1, H2, etc., the spark plugs; T, the timer; MP, the master primary; V, the vibrator; W, the common primary connection; 1, 2, etc., the stationary contacts of the timer. The primary windings are all united in parallel at the top by a wire W, and with the lower ends connecting respectively with the segments of the timer E. The primary winding MP, which operates the vibrator V, is in series with this winding, the wire WT, connecting from the battery and passing directly through the master primary MP. The four condensers C1, C2, C3 and C4, are in parallel with the primary windings. Each of the secondary windings S connects direct to the spark plugs, designated respectively H1, H2, H3 and H4.

On cranking the engine the spark should be "fat" if everything be in good condition; if a weak spark be produced it may be due to either a loose terminal, run down battery, or badly adjusted vibrator. When no spark can be obtained the entire system must be examined and tested, beginning at the battery.

Plug Testing in Multi-Cylinder Engines.—All nuts are removed from the plug, leaving the high tension wires in place. After starting the engine, all wires are grounded except one, thus running the engine on one cylinder. In case there be no misfiring after testing at various engine speeds, it can be taken for granted that the plug is sound. The remaining

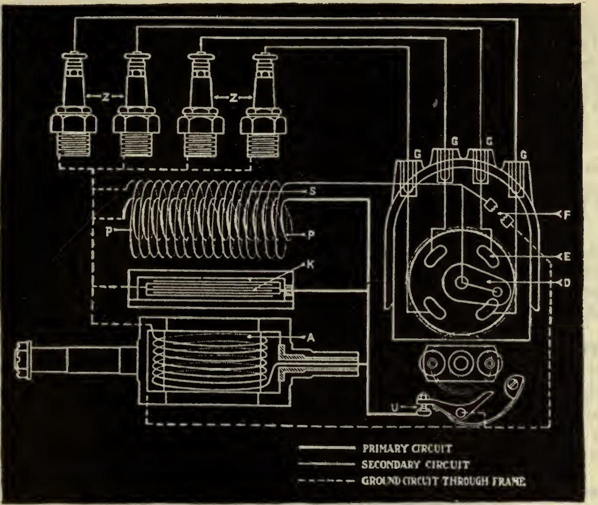


FIG. 2,784.—Circuit diagram of a ***magneto*** with ***self-contained coil.*** A, is the armature winding; P, primary —, S, secondary of transformer; D, distributing brush carrier; E, contact segments; F, safety spark gap; G, terminals to plugs; U, interrupter; Z, spark plugs. ***In operation,*** alternating current flows from the armature having two points of maximum pressure in each armature revolution. As the current leaves the armature, it is offered two paths: 1, the shorter through the interrupter U, to the ground, and 2, the longer through the primary P, of the induction coil to the ground. A third path through the condenser K, is only apparently available; it is obstructed by the refusal of the condenser to permit the passage of the current, as the condenser will merely absorb a certain amount of current at the proper moment, that is at the instant of the opening of the interrupter. The interrupter being closed the greater part of the time, allows the primary current to avail itself of the short path it offers. At the instant at which the greatest current intensity exists in the armature, the interrupter is opened mechanically so that the primary current has no choice but must take the path through the primary P, of the induction coil. A certain amount of current is at this instant also absorbed by the condenser K. This sudden rush of current into the primary P, of the induction coil, induces a high tension current in the secondary winding S, of the coil which has sufficient pressure to bridge the air gap of the spark plug. The sharper the rush of current into the primary winding P, the more easily will the necessary intensity of current for a jump spark be induced in the secondary winding S. ***The distribution of the current*** in proper sequence to the various engine cylinders is accomplished as follows: the high tension current induced in the secondary S, of the induction coil is delivered to a distributing brush carrier D, that rotates in the magneto at half the speed of the crank shaft of the engine. This brush carrier slides over insulated metal segments E, there being one for each cylinder. Each of these segments E, connects with one of the terminal sockets that are connected by cable with the spark plugs as shown. ***At the instant of interruption*** of the primary current, the distributing brush is in contact with one of the metal segments E, and so completes a circuit to that spark plug connected with this segment. Should the circuit between the terminal G, and its spark plug be broken, or the resistance of the spark plug be too great to permit a spark to jump, then the current might rise to an intensity sufficient to destroy the induction coil. To prevent this what is known as a safety spark gap is introduced. This will allow the current to rise only to a certain maximum, after which discharges will take place through this gap. ***In construction,*** the spark discharges over this gap are visible through a small glass window conveniently located.

plugs are tested in the same manner. When a multi-unit coil is used, a faulty plug may be located by holding down all the vibrator blades but one, so that only one spark plug operates. Running each cylinder separately by this means, it can easily be ascertained which plug is defective. Some coils are provided with little knobs for cutting out cylinders in the manner just described.

Complete Break in the Wiring.—The engine is placed upon the sparking point, the primary switch closed, and the two terminals of the suspected wire touched with a test wire. A flow of current indicates a break.

Partial Break in the Wiring.—A partial break, or one held together by the insulation, may sometimes be located by bending the wire sharply at successive points along its length, the engine being at the sparking point and the switch closed as before.

Primary Short Circuits.—The primary wires should be disconnected from the coil, leaving the ends out of contact with anything. There is a short circuit if on touching the switch points momentarily a spark appear. A short circuit may sometimes be overcome by clearing all wires of contact with metallic bodies, and pulling each wire away from the others which were formerly in contact with it.

Secondary Short Circuits.—The secondary lead from the spark plug should be disconnected. Under this condition the high tension current may sometimes be heard or seen discharging from the secondary wire to some metallic portion of the car. Water in contact with the secondary wire will sometimes cause a short circuit unless the insulation be of the best quality.

The Primary Switch.—This portion of the primary circuit sometimes causes trouble by making poor contact. This is generally due to the deterioration of the spring portion of the metal, which gradually loses its resiliency. Snap switches sometimes fail through the weakening of the springs which hold them in the "on" or "off" position. The contacts of a switch should be kept in good condition.

Primary Connections.—All binding posts and their connections should be clean and bright. The wires should be firmly secured to the binding posts, as a loose connection in the primary circuit is often the cause of irregular misfiring or the stopping of the engine.

Vibration.—Since the wires are subject to constant vibration, a number of strands of fine wire is better than a single heavy wire, as the latter is more liable to be broken. In securing the wire to a binding post, care should be taken that all the strands are bound.

Timers.—These may give trouble by: 1, presence of dirt; 2, loose contacts; or 3, division of the spark; this latter effect is sometimes caused by metallic particles wearing off the revolving part forming a path so that the spark passes from the revolving part to more than one contact segment.

Coils.—The part of a coil which requires most frequent attention is the vibrator. The contact points are subject to deterioration on account of the small spark always present between the points when the coil is in operation. In time, the points become corroded and burned, and therefore require to be resurfaced by smoothing with a fine file. A faulty connection to the condenser is at once shown by the large sparks at the vibrator points. Any repairs to a coil, aside from the vibrator, should be done by an expert, as the construction is very delicate.

Igniters.—In make and break ignition, a failure to get a spark, especially with a weak battery, is frequently due to the tappet spring. This spring must be quite stiff so as to cause the break to take place with considerable rapidity; ***the more rapid the break the better is the quality of the spark.*** The contact points of the igniter electrodes are subject to corrosion and wear. When they become pitted the contact surfaces should be filed smooth.

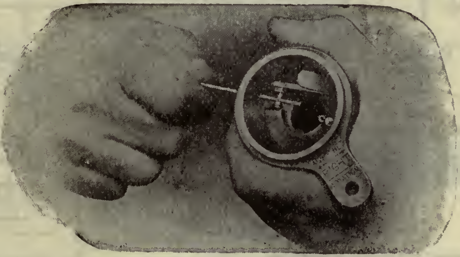
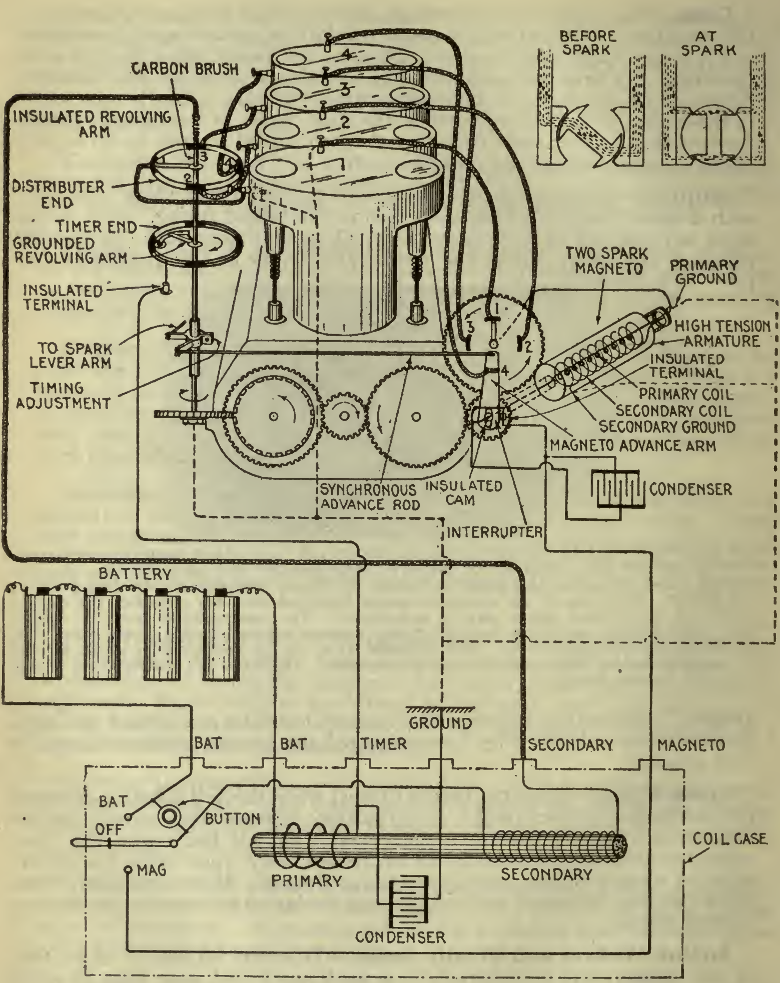


FIG. 2,785.—Filing slot for cleaning platinum contacts of Connecticut magneto interrupter. The cup holding the interrupter or primary circuit breaker may be withdrawn from its housing. The slot serves as a guide for a small flat file for cleaning and squaring the contact points. By means of an adjustable gauge furnished with the magneto, the correct opening of the contact points may be determined. The interrupter is provided with a single roller bearing against the cam pins, thus insuring accurate timing at any speed and unaffected by centrifugal force. The advance lever can be connected at either side of the magneto as the interrupter housing is reversible. The cams are renewable by a half turn with a screw driver.

Spark Plugs.—Repeated failure to start when the coil vibrator operates indicates a faulty spark plug. A rich gasoline mixture often leaves a carbon deposit, and being a partial conductor short circuits the plug. The porcelain insulation, on account of its brittleness, may crack inside the sleeve, allowing a spark to pass there instead of at the gap. Mica insulation sometimes becomes saturated with oil, causing the layers to separate, permitting a short circuit.

Engine Misfires and Finally Stops.—This may be due to exhaustion of the battery, and is indicated by a weak spark and very faint vibrator action.

Engine Suddenly Stops.—This is generally caused by a broken wire or loose switch which does not stay closed. In the case of a single cylinder,



FIGS. 2,786 to 2,788.—Double ignition consisting of a two spark high tension magneto system. and a battery synchronous ignition system with engine driven distributer. Fig. 2,786, elementary diagram of connections; fig. 2,787, position of magneto armature just before time of spark; fig. 2,788, position of armature at time of spark.

the broken wire may be either in the primary or secondary circuit; if a multi-cylinder engine, the break is in the primary circuit.

Engine does not Start.—Usually caused by: 1, primary switch not closed; 2, battery weak or exhausted; 3, entire or partial break in wire; 4, loose terminal; 5, moisture on spark plug; 6, fouled plug; 7, spark too far retarded or advanced; or, 8, too slow cranking with magneto ignition.

Engine Runs Fitfully.—Frequently results from a partial break in the wiring, especially in the primary circuit.

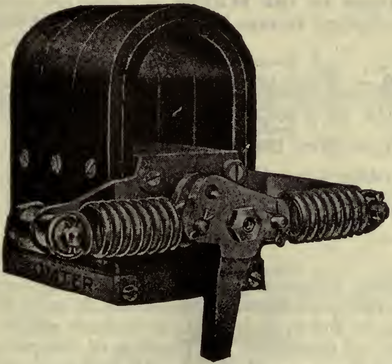


FIG. 2,789.—Sumter low tension oscillating magneto. In this type the armature does not revolve continuously but oscillates back and forth through an angle of about 90°.

Pre-ignition.—Caused by: 1, some small particle in the cylinder becoming heated to incandescence; 2, the electrodes of the spark plug becoming red hot; or, 3, intermittent short circuit in the primary.

Engine Runs With Switch Open.—Usually caused by: 1, overheated engine or plug points; 2, primary short circuit; or, 3, defective switch; 4, an incandescent particle inside the cylinder.

Engine Misfires.—This may be caused by: 1, weak battery; 2, partial break in conductor; 3, loose or disconnected terminal; 4, intermittent short circuit in the secondary; 5, faulty action of either timer or vibrator contacts; 6, bent vibrator blade; 7, faulty spark plug; or, 8, air gap too large.

Knocking of Engine.—Too much advance of the spark sometimes produces this effect.

Knocking in the Cylinder.—The form of unusual noise commonly described as "knocking" consists of a regular and continuous tapping in the cylinder, which is so unlike any sound usual and normal to operation that, once heard, it cannot be mistaken. Too much advance of the spark sometimes produces this result. As mentioned by numerous authorities, the placing of the spark plug in the exact center of the combustion space occasions a peculiarly sharp knock, which may be stopped by advancing or retarding the spark from the one point of trouble. This explanation of the trouble is questioned by others, and is probably overrated.

Loss of Power Without Misfires.—This may be due to badly adjusted coil contacts, poor spark, or incorrect timing.

Explosions in the Muffler.—These are usually caused by misfiring, partially charged storage battery, or by one cylinder not working.

Timing.—The expression "timing the valves" of a gas engine corresponds to "setting" the valve of a steam engine. Evidently the valves must open and close at precisely the proper moment, otherwise the sequence of events of the working cycle would be disturbed and the engine would not operate satisfactorily.

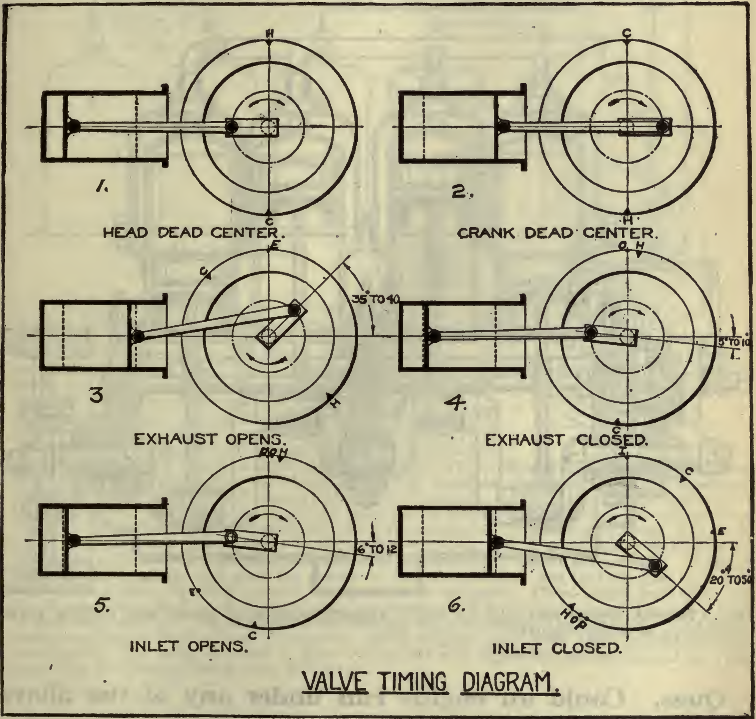
The points of opening and closing of valves are designated in two ways: either in terms of degrees around the fly wheel, or as distance moved by the piston in the cylinder. As it is much easier, after the motor has been assembled, to determine the position of the piston from marks on the fly wheel, the former method for setting valves has been almost universally adopted.

As soon as the engine is finished, two marks, diametrically opposite, are located on the rim of the fly wheel, such that when one is directly over the center of the main shaft, the piston will be at one end of its stroke, or in other words, when either of these marks is on top, the piston will be on one of its "dead centers."

In figs. 2,790 to 2,795, at 1, an engine is shown at the head center, and at 2, the same engine is shown at the crank center, the rotation being indicated by the arrow. At 3, the position of the crank at exhaust opening is shown, which varies from 35° to 40° from the bottom center of the power stroke in various engines. At 4, the exhaust closes about 5°–10° beyond the top center of the exhaust stroke. At 5, the inlet opens about two degrees after the closing of the exhaust, and at 6, the inlet valve closes and compression commences. Now, if by turning the engine over, the exhaust valves be found to seat at or before the dead center, it is plain that it closes 5°–10° too early, and must have opened the same amount too early. Also, if the inlet valve should be found to open on or before the top center, it must also have closed too early, and thus prevented the engine taking a full charge.

Ques. Is the operation of the exhaust and inlet valves dependent upon each other?

Ans. If all of the valves be operated by the same cam shaft, the valves are bound to be "in step" with each other, providing there is no excessive lost motion, but if one of the valves be early or late, it is evident that they are all out of time.



FIGS. 2,790 to 2,795.—Valve timing diagrams, showing timing of valves for one cylinder engine.

Ques. What might happen if the exhaust and inlet valves be operated by separate cam shafts?

Ans. Both sets of valves may be out of time with each other, and at the same time being out of time with the pistons, or one set may be in time with the pistons, while the other set is out of time.

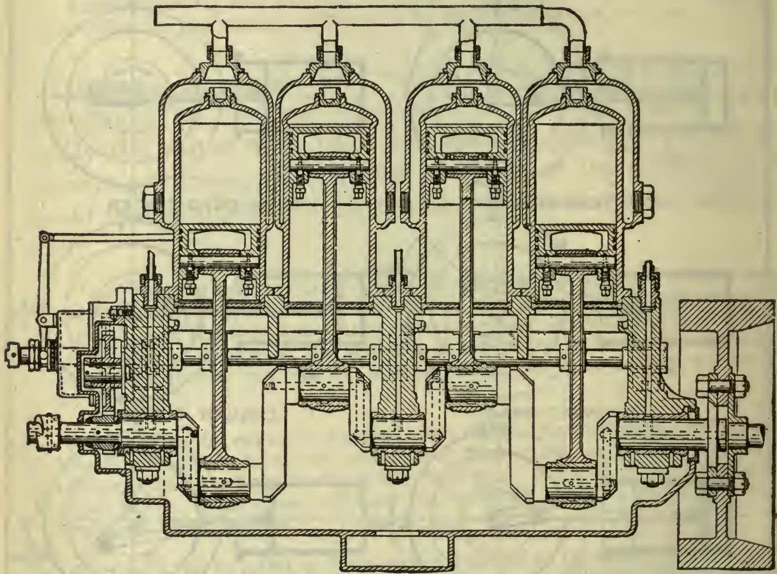


FIG. 2,796.—A typical four cylinder engine showing position of cranks and working parts, including secondary shafts.

Ques. Could an engine run under any of the above named conditions?

Ans. The valves may all be considerably out of time, and the engine will still run but at the expense of reduced power, increased fuel consumption, and excessive vibration.

Ques. How is improper timing detected?

Ans. By unsteady action, especially at the higher speeds.

Ques. If the valves be out of time, and there be no lost motion, what must be done?

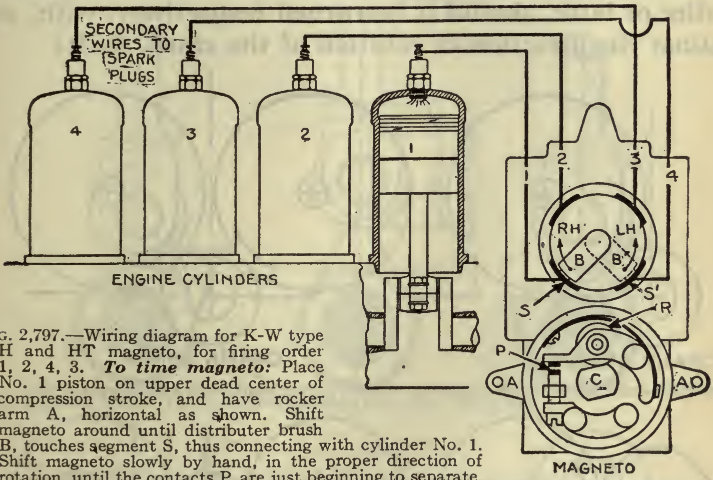
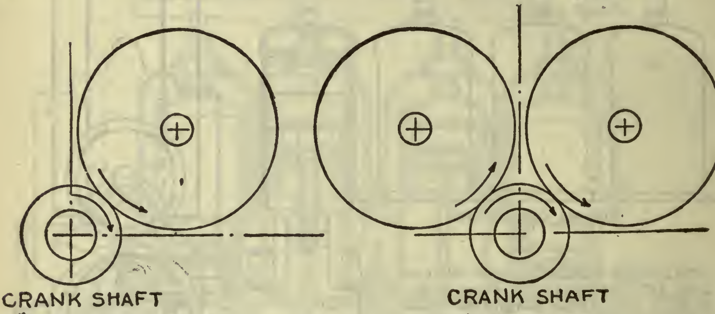


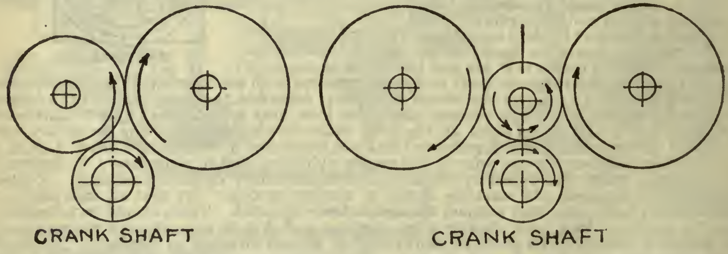
FIG. 2,797.—Wiring diagram for K-W type H and HT magneto, for firing order 1, 2, 4, 3. ***To time magneto:*** Place No. 1 piston on upper dead center of compression stroke, and have rocker arm A, horizontal as shown. Shift magneto around until distributer brush B, touches segment S, thus connecting with cylinder No. 1. Shift magneto slowly by hand, in the proper direction of rotation, until the contacts P, are just beginning to separate. At this point secure magneto shaft to gear or coupling with set screws. When one cylinder is timed, proceed to connect the others as follows: Ascertain the firing order of the engine, then crank engine slowly and connect plug cable from next cylinder that fires to distributer segment No. 2 and so on until all the plug cables are connected. The secondary connections on the hard rubber distributer block are numbered in consecutive order, 1, 2, 3, 4, etc. These numbers do not refer to the engine cylinders, and it is necessary to determine the order in which the cylinders fire and connect secondary cables accordingly. Replace parts on the magneto and start the engine to test the setting. See that all nuts and connections are tight, also that retainer spring has been replaced. There should be a tendency for the engine to kick back slightly when starting, and if it do not, advance magneto until it does kick slightly. To advance, shift coupling against direction of rotation. To retard, shift coupling with direction of rotation. Shift slightly each time until correct position is obtained. Pin magneto shaft to gear or coupling with taper pin, do not depend on a set screw, as it will surely work loose in time.

Ans. One of the pistons, say, 1, in fig. 2,797 is put on top center, a space amounting to about 5° to 10° is then marked off on the fly wheel rim, and the engine turned ahead until this mark comes even with the center pointer; the gears of the exhaust cam shaft are then shifted so that the exhaust cams just allow the valves to remain clear of their seats. The engine is then turned ahead about 2°, and the inlet valve shaft shifted until the inlet valve commences to raise.

Ques. When shifting the cam shaft to make the valves earlier or later, should it be turned respectively, with, or against the direction of rotation of the crank shaft?



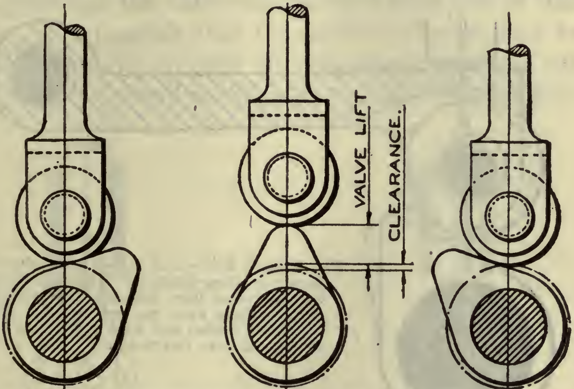
FIGS. 2,798 and 2,799.—Diagram showing rotation of cam shafts with a direct gear device.



FIGS. 2,800 and 2,801.—Diagram showing rotation of cam shafts when an intermediate idler is used.

Ans. That depends upon the gearing. If the cam shaft gears mesh directly into the crank shaft pinion, the cam shafts revolve in an opposite direction than the crank shaft, and to make the valves earlier, the cams must be turned in the direction followed by the cam shafts, which is opposite that of the engine.

Most engines have an idler gear between the crank shaft and the cam shaft gears; in such case the cam shafts revolve in the same direction as the crank shaft, and the cams shift ***with*** or ***opposite*** the engine to make them respectively early or late. This is illustrated in figs. 2,798 to 2,801.



FIGS. 2,802 to 2,804.—The successive positions of a valve lifting cam.

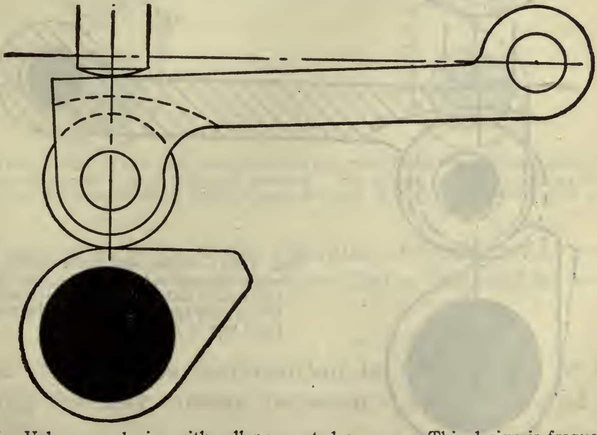


FIG. 2,805.—Valve cam device with roller mounted on cam. This design is frequently used to reduce the side thrust upon the push rod.

Ques. If all valves be driven from one cam shaft, is it possible that one or more of them may be timed correctly, while the others are out of time?

Ans. On some engines the cams are made independent from the cam shaft, and are keyed to the latter, hence, it is possible

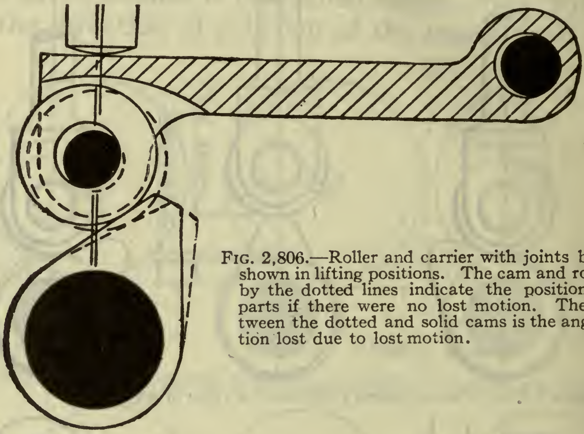


FIG. 2,806.—Roller and carrier with joints badly worn shown in lifting positions. The cam and roller shown by the dotted lines indicate the position of these parts if there were no lost motion. The angle between the dotted and solid cams is the angle of rotation lost due to lost motion.

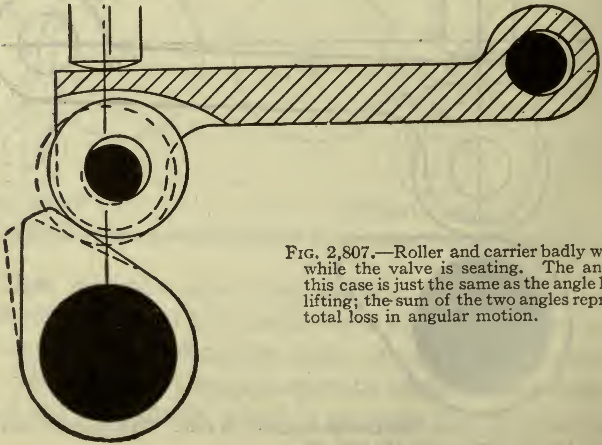
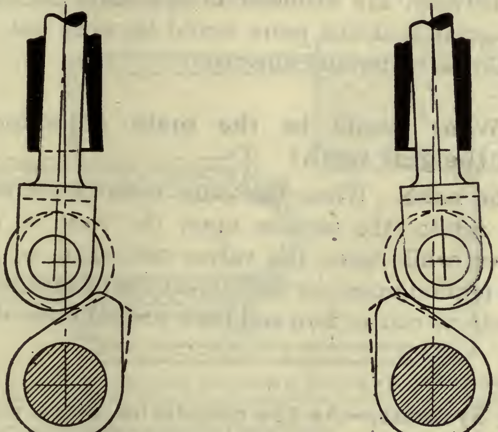


FIG. 2,807.—Roller and carrier badly worn shown while the valve is seating. The angle lost in this case is just the same as the angle lost during lifting; the sum of the two angles represents the total loss in angular motion.

that, in overhauling the engine, some of the cams may have been replaced wrongly.

Ques. Is this possible if the cams be forged with the shaft?

Ans. Although the cams cannot be shifted out of time with each other, it is possible that the surface of some may be worn more than others, or on such engines where the cams act upon rollers carried by the end of the push rod, as in figs. 2,802 to 2,804, or by a carrier as in fig. 2,805, the roller and pin may be worn to such an extent as to make proper timing impossible, without removal of the worn parts.



FIGS. 2,808 and 2,809.—Valve push rod with roller shown in lifting and seating position, showing the serious defects of a worn push rod bushing. The dotted cams and rollers indicate the position these parts would have with the same amount of lift if there were no lost motion.

Ques. Could this lost motion be taken care of when adjusting the lost motion between valve stems and push rods?

Ans. No, between valve stem and push rod only the lost motion in the direction of the valve travel can be adjusted. The lost motion in the rollers, the carrier suspension, or the push rod bushings, acts in a direction at right angles to the valve travel, and affects the timing in the manner shown in figs. 2,806 to 2,808.

Ques. Will wear of the gear teeth affect the timing?

Ans. Very little. even if the teeth be worn considerably, the difference between the diameter of the gears and that of the cams is so great that the gears would be worn out before the timing would be materially affected.

Ques. What would be the main objection to lost motion in the gear teeth?

Ans. The noise. When the cams raise the valves against the spring action, the tension upon the teeth is upon their driving faces, while when the valves seat again by the spring action, the tension upon the teeth is on the reverse side, causing the gear teeth to clatter fore and back several times during each revolution.

Cooling Systems.—As the cylinder of a gas engine is an explosion chamber, that is, a furnace wherein the fuel is burned, and the explosions are very frequent, it is necessary to adopt some means to cool the cylinder walls. If the cylinder were not cooled in some way, it would get red hot, lubrication under such conditions would be impossible, and the casting would be destroyed in a short time.

In addition, the temper would be taken out of the valve springs, the spark plugs would possibly crack, and the incoming charge become so rarefied as to seriously impair the power of the engine.

Although a cooling system is a necessity, for the reasons stated above, it always causes a loss by absorbing a part of the heat units generated by the combustion of the fuel, thus reducing the efficiency of the engine.

The two methods used to cool the cylinder are

1. A jacket of circulating water;
2. Induced air currents.

In the first method a thin space around the cylinder is provided for the water by an outer casing of metal, either cast with the cylinder, or attached as in the case of a sheet copper casing.

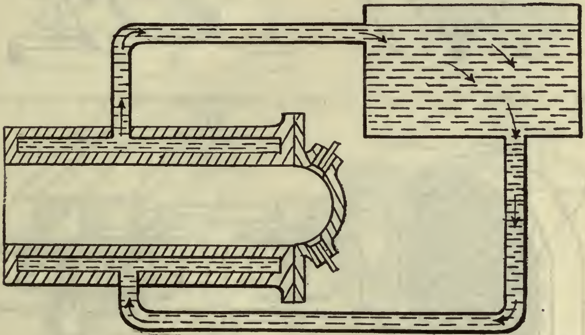


FIG. 2,810.—Diagram of a gravity water circulation system for a gas engine cylinder. As indicated by the arrows, the water from the tank enters the jacket of the cylinder at the lowest point, and being there subjected to the heat of the cylinder walls, rises to the level of the tank water; thus maintaining a continuous circulation.

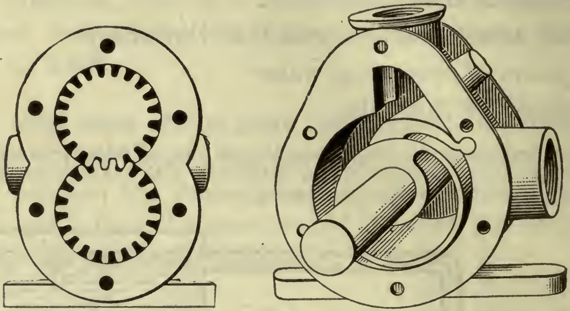
The cooling or *circulating water*, after passing through the jacket and absorbing heat from the cylinder, passes off to 1, a tank, 2, radiator (in the case of an automobile), or 3, to waste. In the first two cases the water is used over and over again.

Since the heat causes the water to evaporate, and in some cases, to boil, it must be replenished from time to time where a tank or radiator is used.

In tank or radiator systems the water is kept in circulation by

1. Gravity, or
2. Mechanical means.

Fig. 2,810 shows the gravity method. The water, expanding with rise of temperature becomes lighter and the heated layers rise while the cool layers descend, thus a continuous circulation is produced. In the mechanical method the water is kept in circulation by some form of pump operated by the engine.



FIGS. 2,811 and 2,812.—Two types of circulating pump for use in the water cooling system of gas engines.

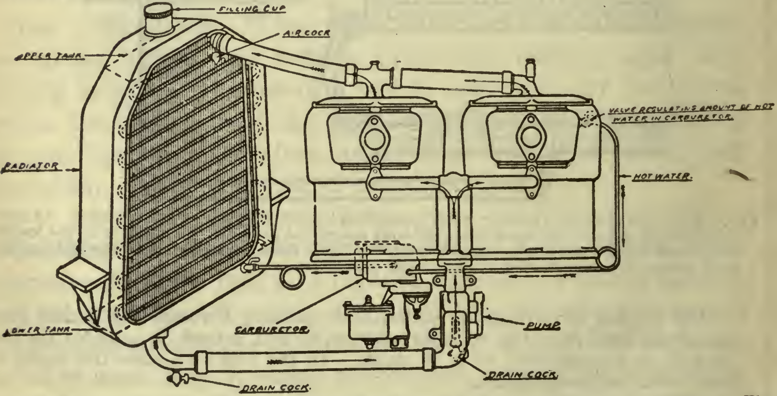
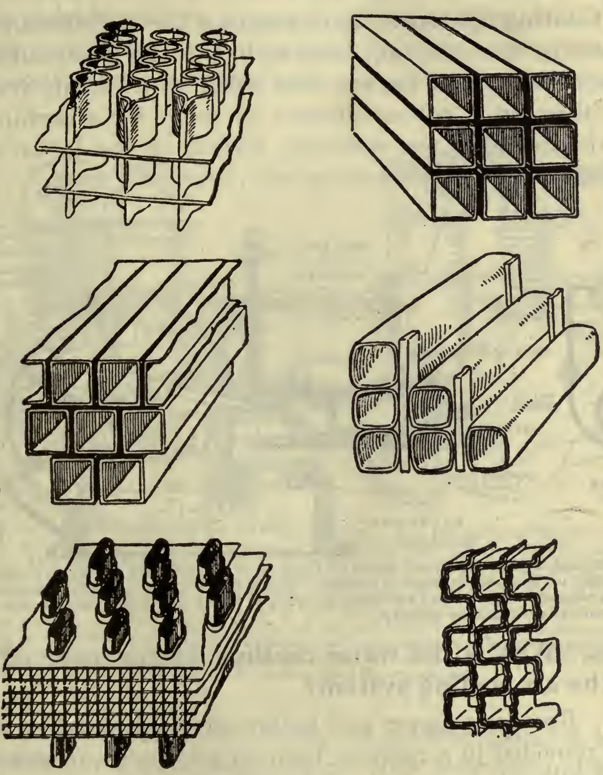


FIG. 2,813.—An example of a radiator and water cooling system with pump circulation. The cooling is assisted by a fan geared to the engine which induces a current of air through the radiator when the car is standing.

Ques. What is the construction of a "honeycomb" radiator?

Ans. This type consists of numerous short lengths of small tubing of square cross section, placed side by side and held together by solder at the ends, the structure presenting the appearance of a "honeycomb."

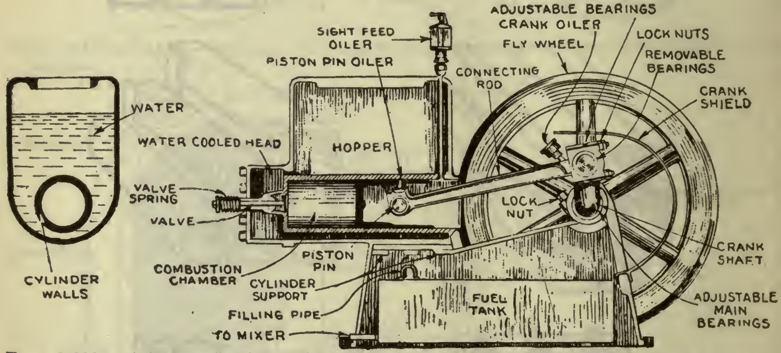
Ques. Explain the operation of a honeycomb radiator?



FIGS. 2,814 to 2,819.—Types of radiator construction; fig. 2,814, Harrison tubular; fig. 2,815, Fedders staggered air tube; fig. 2,816, Fedders square air tube; fig. 2,817, A-Z vertical type; fig. 2,818, Buscoe vertical type; fig. 2,819, Livingston zig-zag tube. ***In the construction*** of radiators, copper, or its alloy, brass, is used on account of its great capacity for conducting heat, the weight then can be kept low. It also has mechanical advantages such as ease of forming and soldering.

Ans. The heated water from the engine enters at the top and circulates to the bottom, flowing through the small spaces between the tubes. A strong current of air induced by a fan, and also by the motion of the car, passes through the tubes and absorbs heat from the water.

Air Cooling System.—In this method the cylinders are cooled by a strong current of air, induced in the case of an automobile by the movement of the car, and aided by a rapidly revolving fan. Since air is not as efficient as water for absorbing heat from the cylinder, the radiating surface of the latter is considerably increased by numerous ribs or rings.



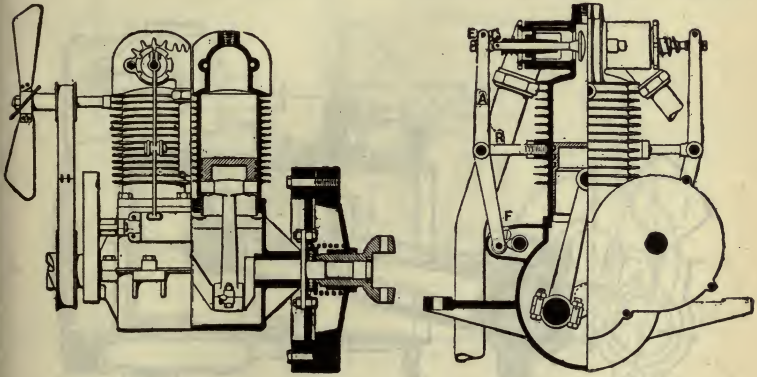
FIGS. 2,820 and 2,821.—Sectional views of Fuller and Johnson gas engine cylinder showing open jacket or "hopper" method of cooling. The hopper consists of a large vessel surrounding the cylinder, which holds a sufficient quantity of water to adequately cool the cylinder. The water circulation is by gravity.

Ques. Why is the water cooling system more efficient than the air cooling system?

Ans. Because a larger and better arranged radiating surface can be provided in a radiator than on a cylinder; in either case the excess heat is carried off by the air. The water simply serves as a medium to conduct the heat from the engine to the radiator.

Lubrication Systems.—There are several methods of gas engine lubrication in general use, and they are known as gravity, splash, pressure, and positive systems.

In the ***gravity system,*** the lubricator is placed at a sufficiently high elevation to permit the oil to flow to the bearings; 2, in the ***splash system,*** a quantity of oil is placed in the crank case and maintained at such a level that the ends of the connecting rods come in contact with the oil at the lower part of their revolution and splash it upon the working parts; 3, in the ***pressure system,*** the oil is contained in a reservoir and forced to the various bearings under pressure acquired by connecting the reservoir to the exhaust by a small pipe, or by utilizing the pressure from an enclosed crank case; in the ***positive system,*** a



FIGS. 2,822 and 2,823.—The Cameron air cooled engine. The fan shown at the left induces a current of air which passing over the large surface presented by numerous ribs, cools the cylinders. The valves are located above the cylinder bore in opposite chambers and work horizontally. Each valve is operated by a long vertical lever A, pivoted at R. The upper end C, bears upon the end of the valve stem and its lower end carries a roller against which bears the camshaft cam D. The upper end of the lever or valve rocker arm is split and takes a threaded piece E, which rests upon the end of the valve stem. By the adjustment of this the timing of the valve is accomplished. The lower end, with its roller is contained within a small extension on a detachable plate secured to the side of the crank case and the end of the valve rocker arm working in a slot F, in the top of the expansion.

pump geared to the engine forces a certain amount of oil through the feed at each stroke of the plunger.

Ques. How is a gas engine cylinder lubricated?

Ans. In some engines the splash system is used, while in others the oil is fed from a lubricator.

Ques. What determines the choice of a lubricant?

Ans. The principal things to be considered are: 1, rubbing pressure; 2, rubbing velocity, and 3, temperature.

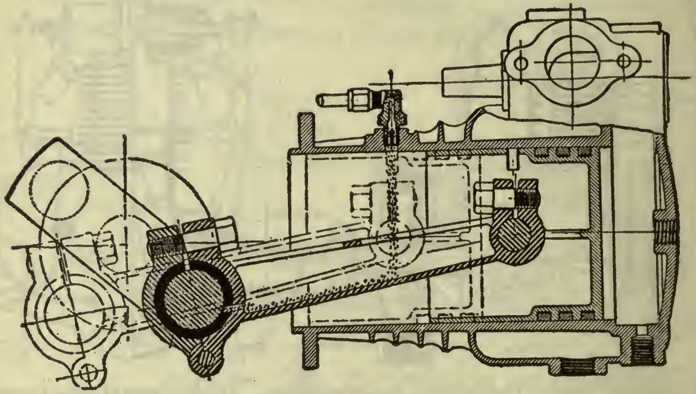


FIG. 2,824.—Horizontal cylinder oiled by force feed oiler distributer. The piston is oiled whe passing under oil port, as shown by the dotted outline. The connecting rod is longitudinally grooved on the upper surface, so as to carry oil to the bearings.

Ques. How should oil be selected for cylinder lubrication?

Ans. It is desirable to select an oil that will deposit as littl carbon as possible.

Ques. Are animal oils suitable for engine lubrication

Ans. Animal oils, such as sperm, whale, fish, lard and neat' foot oils are sometimes used on outside bearings of heavie machines, but for high speed machinery especially gas engines, with the accompanying high temperature, they should not be used.

Ques. What kind of oil is generally used for engine lubrication?

Ans. Mineral oil of considerable body yet of high fluidity and cold point.

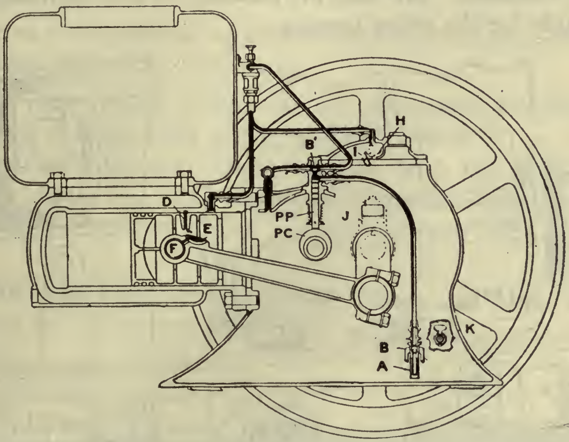


FIG. 2,825.—Gray engine lubrication system. The oil is poured into the reservoir in the bottom of the engine through the cap opening H, in the top of the crank case. The pump PP, forces oil to all working surfaces after drawing it through the strainer A, at bottom of base. A large pocket J, is located directly over each crank shaft bearing (these pockets can be seen by looking at the outside of the engine) and catches oil from the feed pipe and keeps the main bearings bathed in oil at all times.

Ques. What qualifications must a cylinder oil possess?

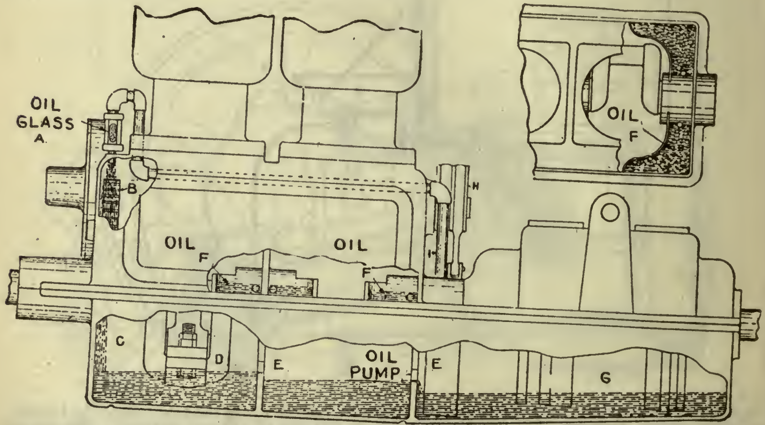
Ans. It should have a "flash" point of not less than 360° Fahr., and a fire test of at least 420°, together with a specific gravity of 25.8, and a viscosity of 175.

Ques. Why must the oil have a high flash point?

Ans. As the piston rises in the cylinder, the oil is deposited on the walls, when the piston moves outward the oil is exposed to the heat of the burning gases. The length of time during which it will continue to lubricate under this condition will determine the value of the oil.

Ques. Is it customary to use any other than mineral oil on the bearings when a common lubricator is used?

Ans. Only one grade of oil is used, and this is selected to suit the cylinders. Oils that are suitable for gas engine cylinders are suitable for the other bearings.



FIGS. 2,827 and 2,827.—Palmer marine engine lubrication system. The pump forces oil from the base reservoir through the pipe I, and oil glass A, falling over gears B, and forward crank bearings. At C, oil is shown dropping from gears and crank bearings to the forward compartment, oiling crank pin bearing D, and splashing by throw of connecting rod to cylinder walls and wrist pin. The centrifugal motion of the horizontal gears throws the oil that drips off the large cam gear into the cam bearing. The opposite forward cam shaft bearing has a recess to catch the oil, supplying the bearings. Partition plates and oil passages maintain a uniform level of the oil. The circulation is continuous.

Engine Management.—This includes, not only the necessary conditions of operation and control, but also the numerous mishaps that may be encountered, as those arising:

1. From faulty construction, which, however will be seldom experienced with well made engines.

2. From careless or ignorant handling, such as:

   a. Insufficient lubrication;
   b. Faulty adjustments;
   c. Exhaustion of the fuel, ignition current, or jacket water;
   d. Racing;
   e. Over heating.

3. From any one of a number of disorders in the ignition apparatus.

4. From poor gasoline, or faulty adjustment of the carburetter.

5. From worn or broken parts.

By far the greater proportion of gas engine troubles result from some derangement of the ignition system.

Second in importance come troubles with the fuel mixture.

Again, either or both electrical apparatus and carburetter may require attention.

Before Starting the Engine.—There are three supplies necessary for the operation of a gas engine:

1. Fuel;
2. Lubricating oil;
3. Circulating or cooling water.*

In filling the gasoline tank, the liquid should always be strained to guard against the carburetter passages becoming clogged by any foreign matter that may be contained in the fuel. A chamois skin or wire netting with very fine mesh should be used as a filter.

After filling the tank, the filler cap should be replaced and care taken that the small hole in the center of the cap is open so that air may be admitted as the fuel is used, to prevent the pressure within the tank becoming less than that of the atmosphere.

The valve on the fuel supply pipe should now be opened and after sufficient time has elapsed for the float chamber of the carburetter to fill.

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*NOTE—Except in case of air cooled engines.

Next, the circulating water tank should be filled with *clean water*. Care should be taken with the water, to see that it is free from any foreign matter so as to guard against the passages of the cooling system becoming obstructed.

Before starting the engine, all the working parts requiring lubrication should receive attention and in all particulars it is well to adhere to the manufacturer's instructions in the performance of this task.

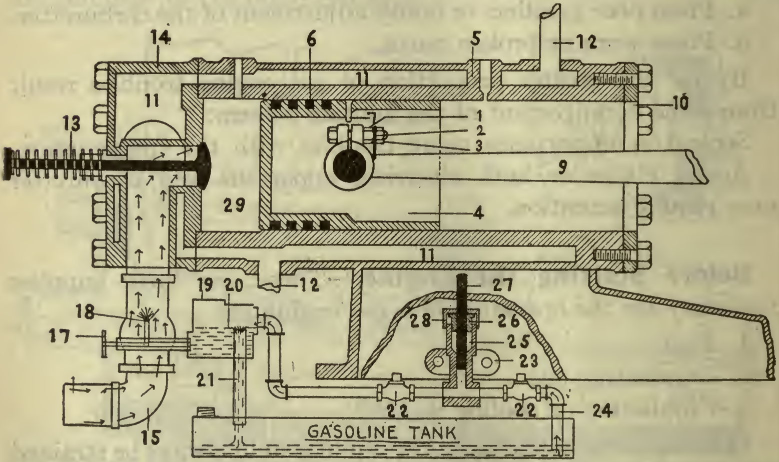


FIG. 2,828.—Sectional view of Gilson four cycle gasoline engine cylinder. ***The parts are:*** 1, adjusting nut; 2, lock washers; 3, jam nuts; 4, piston; 5, cylinder; 6, piston rings; 7, piston bearing oil tube; 8, piston bearing; 9, connecting rod; 10, cylinder packing ring; 11, water jackets; 12, water inlet and outlet; 13, intake valve; 14, cylinder head; 15, intake pipe; 16, gasoline needle valve; 17, gasoline feed pipe; 18, gasoline reservoir; 19, gasoline level; 20, overflow pipe; 21, check valves; 22, gasoline pump; 23, gasoline suction pipe; 24, pump packing; 25, packing gland; 26, pump plunger; 27, packing gland regulating nut.

The quality of lubricating oil required for gas engine cylinders is quite different from that used for steam engines. Owing to the high cylinder temperatures a gas engine must have an oil possessing a high fire test. As the average cylinder temperatures may be said to be from 300 to 400 degrees Fahrenheit, an oil should be used having a fire test higher than the latter figure; the flashing point should not be less than 400 degrees. Air cooled engines, being hotter under working conditions than water cooled, require a lubricant capable of withstanding higher temperatures than that required by the latter.

In addition to the attention required in supplying gasoline, water and oil as just described, it is necessary before starting the engine to make sure:

1. That all parts of the lubricating system are in working order, all connections opened, and the supply of oil sufficient;

2. That the ignition circuit is closed, which involves examination of all switches, to insure certainty that they are on the "closed" point;

3. That the carburetter control levers be placed in position for ensuring the *richest* mixture under operating conditions, in order that, even with the low suction at starting, sufficient power may be obtained for a good headway. A rich mixture may occasionally fail to ignite at starting, but a weak mixture is more often at fault.

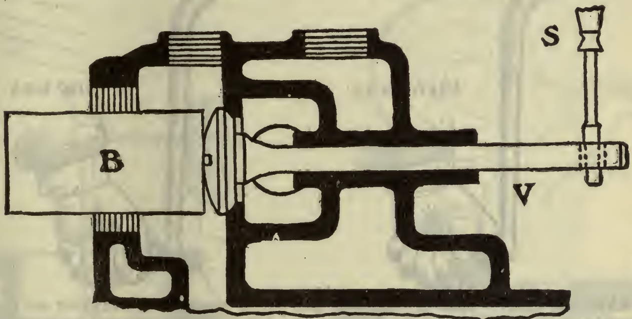


FIG. 2,829.—Method of grinding valves in horizontal cylinders. A block of steel B, is held against the head of the valve V, and the latter rotated on its seat by means of a screw driver blade S inserted in the slot in the stem, the face having been previously trued by a truing tool. In cases where the stem of the valve has no slot, a pair of gas pliers can be used to grip it, being careful in so doing not to mutilate the threads thereon. The valve may be turned by a piece of steel wire inserted in the pin hole.

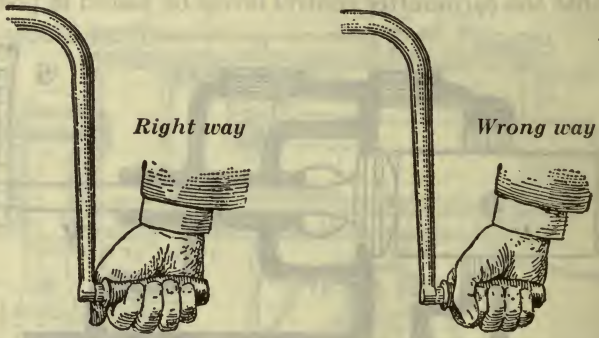
4. That the spark is fully retarded. *To neglect this may cause "back kick" at cranking, and possibly result in serious injury to the operator.*

Spark and Throttle Adjustments before Starting.—On account of the slow speed at which the engine is turned over in *cranking*, it is necessary that the throttle have a considerable degree of opening, and that the spark be fully retarded because of:

1. The weak suction of the piston at slow speed;

2. The need of ensuring a mixture that will ignite under such conditions;

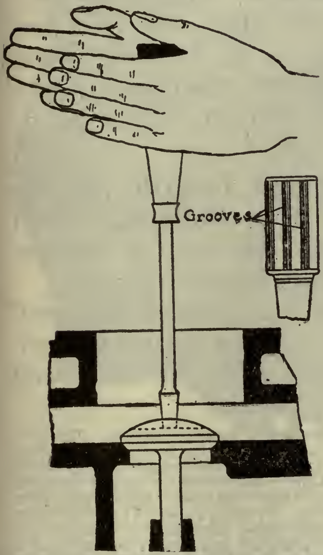
3. The danger of bodily injury from a "back kick" of the engine, which is liable to occur with an early spark at slow speeds.

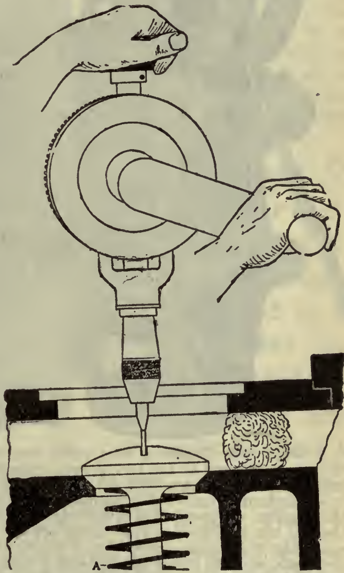


FIGS. 2,830 and 2,831.—Illustrating right and wrong methods of cranking an engine. As ordinarily practiced, the hand is so placed that the thumb and fingers encircle it. Such a method is decidedly unsafe should the operator press down on the crank and a back fire occur. The correct method is to place the thumb on the same side of the handle that the fingers are placed, so that the handle is not entirely encircled, allowing the handle to slip out of the grasp when it is being pressed down, and permitting the fingers to release the handle if it is being pulled up, at the time of the back fire. ***In cranking,*** the operator should begin by ***pulling up, not by pushing down***, as is almost universally done; doing it the latter way ***indicates either ignorance*** or disregard for personal safety.

Starting the Engine.—If the engine can be turned over by hand, as is mostly the case with all engines below twenty-five horse power, and a great many up to fifty horse power, turn the fly wheel backward until it cushions, then forward two or three quick turns, all the way around, and the engine will start.

If the engine is to be started by compressed air, open the air cocks, at the tanks, so as to fill the pipe leading to the engine with compressed air. When gas is used for fuel, open the gas valve, and immediately thereafter, the compressed air cock at that cylinder which is temporarily used to run the engine with compressed air. As soon as good explosions occur in the other cylinders, shut off the air cock, throw in the inlet cam clutch of the starting cylinder, so that it will admit gas for its own explosions. Care should be taken to shut off the air cock before





Figs. 2,832 and 2,833.—Method of using a screw driver for valve grinding. A handful of waste or a cloth is put in the valve port to protect the cylinder, and the valve face coated with a paste of the emery powder and oil and put in place. The handle of the screw driver is now held between the palms of the hands as in the sketch, and a series of oscillations through a small arc given to the valve by moving the palms in opposite direction. After about thirty of these oscillations have been given, the valve is lifted from its seat, given a half turn, and reseated for further grinding in the same manner. This operation should be continued with occasional additions of oil and emery, until the valve face and the seat appear to be bright for their full width around the circle. Fig. 2,833 shows a good form of handle for use in grinding.

Fig. 2,834.—Method of grinding a valve with a drill stock. A screw driver bit is inserted in the chuck and the operation conducted as in the case where a screw driver is used. The crank should be rocked through a small arc instead of being rotated. The spring A, is fitted within the valve chamber to unseat the valve when it is desired to examine it.

the inlet cam clutch of the starting cylinder is thrown in, otherwise this cylinder may compress a charge of air to such a high pressure that it will reverse the piston with disastrous consequences. If the ignition current be obtained from a dynamo or magneto, turn the current on from the dynamo, and cut out the battery from the spark coil circuit.

Pump up and lock the air tanks at the pressure required to start the engine. The pressure varies according to the type and horse power of the engine, and ranges from 100 to 250 lbs.

Spark and Throttle Adjustments after Starting.—When the engine has speeded up, the adjustments must be changed:

1. The spark must be advanced.
2. The throttle opening reduced.

On account of the spark and throttle adjustments necessary before starting, the engine when started will begin to race unless it be fitted with a governor, hence the operator should immediately reduce the throttle opening and advance the spark. If there be a mechanical governor on the engine, the throttle will shut down automatically, as the engine speeds up.

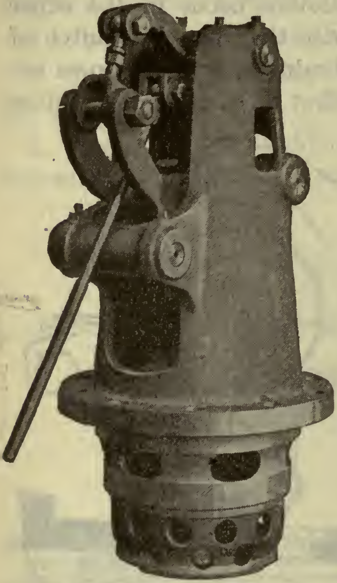


FIG. 2,835.—Allis-Chalmers inlet valve and bonnet assembled of large gas engine. The inlet valves are located at the top of each cylinder, the exhaust valve at the bottom. Each inlet and exhaust valve, together with its seat, guides, operating levers and spring is self-contained in a substantial cast iron chamber which may be readily removed for convenient access to these parts. A special rig is included for lowering the exhaust chambers and swinging them clear of the cylinders. The valves are of the mushroom or poppet type. The inlet valves are made of a special grade of cast iron with forged steel stems. Exhaust valves are cast in one piece and are cored out for water cooling. The valves are operated by eccentrics on a lay shaft driven through spiral gears from the crank shaft, the throw of the eccentrics being transmitted to the valves through rods and steel rolling levers.

Failure to Start.—Refusal of the engine to take up the cycle, even after prolonged cranking, is a familiar experience.

Unless some accident has occurred, or a very unusual strain has been thrown upon the working parts, the inference is that some element of the complicated group of mechanisms is out of adjustment. In most cases the failure to start is due to some faulty adjustment or defect of the carburetter or ignition system.

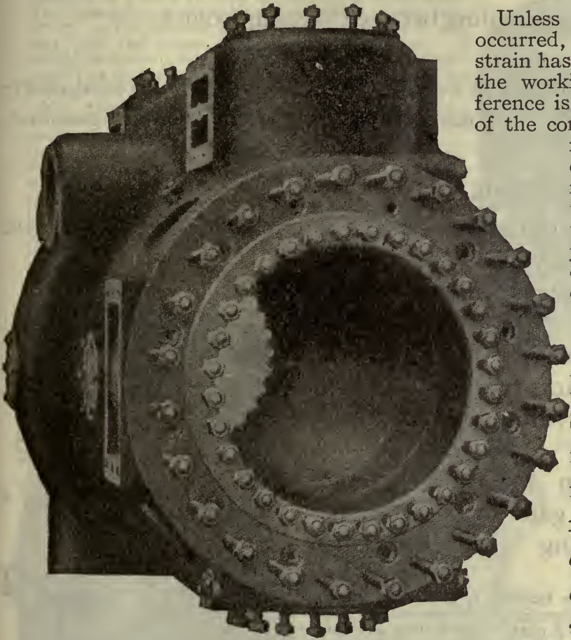


FIG. 2,836.—Allis-Chalmers cylinder for large gas engine. ***In construction,*** it is a one piece casting of special iron with water jackets, mixing chamber and gas and air belts cast integral. This construction eliminates all unsightly gas and air piping above the floor line. A renewable lever is used for the working barrel. The liner is held in the cylinder by a patented locking device in the center and also by a shrink fit throughout its entire length. Hand holes are provided leading to the water jacket space to permit inspection and cleaning if necessary.

Causes of Failure to Start.—In starting the engine, it should be remembered that *a few rapid turns of the fly wheel will do more toward starting an engine than ten minutes of slow grinding.*

If an engine show good compression and will not start after four or five turns, it is useless to continue. Assuming that all the preliminaries to starting have been carefully observed, the probable causes of trouble should be sought in the ignition system of carburetter.

Defective Spark Plugs.—The engine will not start when:

1. The plug points are too far apart!

2. The plug is short circuited.
3. The insulating layer of porcelain or mica is broken down.
4. There is much fouling between the plug points.

Fouling may consist of oil or soot. Both give trouble at starting. Fouling with soot may generally be removed with gasoline.

Preventives of fouling are:

1. An annular space between the core insulation and the outer shell producing a vortex, as is alleged, and allowing piston suction to remove deposits.

2. An auxiliary spark gap, which will generally suffice to insure a spark, but it does not prevent fouling between the spark points. A temporary gap may be made by disconnecting the lead wire

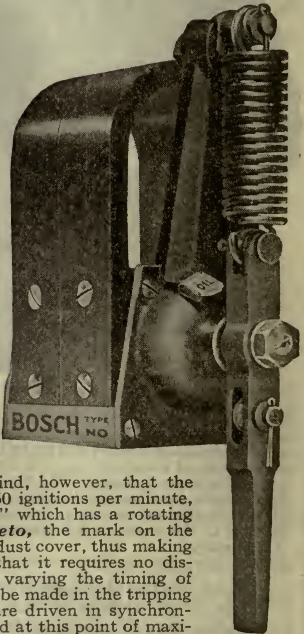


FIG. 2,837.—Bosch low tension, type NO, oscillating magneto used in connection with mechanical make and break igniters. A current wave being produced by each oscillation of the armature, it is necessary to drive the magneto in a fixed relation to the engine crank shaft. The magneto trip lever should be mechanically connected to the movable igniter on the engine, as in this way only can proper synchronism be obtained. It should be borne in mind, however, that the type "NO" cannot be used at speeds greater than 250 ignitions per minute, and where a greater speed is desired, the type "NR" which has a rotating armature, should be used. ***In timing the magneto,*** the mark on the armature should register with the proper mark on the dust cover, thus making the timing for either rotation extremely simple, in that it requires no disassembling of the instrument. Since no method of varying the timing of the spark is provided, arrangements to this end should be made in the tripping mechanism, and since the igniter and the armature are driven in synchronism by the trip lever, the instrument is always operated at this point of maximum efficiency regardless of whether the spark be retarded or advanced. In order to obtain proper results, the trip lever should be deflected through an angle of 30 degrees before it is released, and, since the spark is produced by spring action rather than directly through the speed of the engine and therefore is independent of the latter, no battery is necessary for starting, and in ordinary cases one turn of the fly wheel will be sufficient, provided, of course, that a proper gas mixture be present.

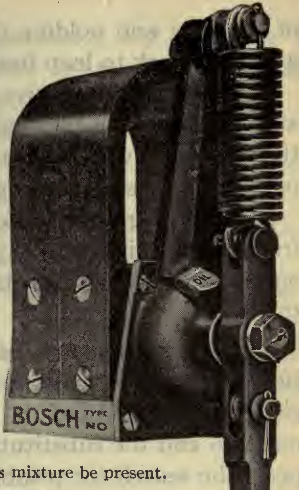


FIG. 2,838.—Bosch low tension, type NO, oscillating magneto used in connection with mechanical make and break igniters. A current wave being produced by each oscillation of the armature, it is necessary to drive the magneto in a fixed relation to the engine crank shaft. The magneto trip lever should be mechanically connected to the movable igniter on the engine, as in this way only can proper synchronism be obtained. It should be borne in mind, however, that the type "NO" cannot be used at speeds greater than 250 ignitions per minute, and where a greater speed is desired, the type "NR" which has a rotating armature, should be used. ***In timing the magneto,*** the mark on the armature should register with the proper mark on the dust cover, thus making the timing for either rotation extremely simple, in that it requires no disassembling of the instrument. Since no method of varying the timing of the spark is provided, arrangements to this end should be made in the tripping mechanism, and since the igniter and the armature are driven in synchronism by the trip lever, the instrument is always operated at its point of maximum efficiency regardless of whether the spark be retarded or advanced. In order to obtain proper results, the trip lever should be deflected through an angle of 30 degrees before it is released, and, since the spark is produced by spring action rather than directly through the speed of the engine and therefore is independent of the latter, no battery is necessary for starting, and in ordinary cases one turn of the fly wheel will be sufficient, provided, of course, that a proper gas mixture be present.

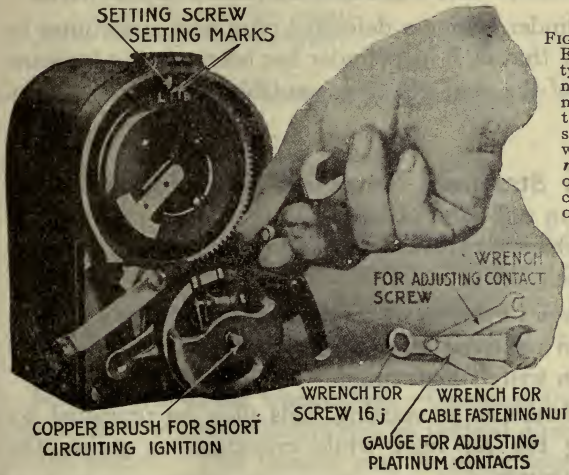

SETTING SCREW
SETTING MARKS
WRENCH FOR ADJUSTING CONTACT SCREW
WRENCH FOR SCREW 16 j
WRENCH FOR CABLE FASTENING NUT
COPPER BRUSH FOR SHORT CIRCUITING IGNITION
GAUGE FOR ADJUSTING PLATINUM CONTACTS

FIG. 2,839.—Eisemann, type G4, magneto showing method of timing and special wrench. ***Timing the magneto;*** As the spark occurs when the primary circuit is broken by the opening of the platinum contacts in the make and break mechanism, it is necessary that the magneto will be so timed that at full retard position of the timing lever body, the platinum contacts will open when the respective piston of the engine has reached the top point on the firing stroke. Turn engine by hand until piston of No. 1 cylinder is on the upper dead center; remove distributor plate from magneto and turn the driving axle of the armature until the setting mark on the distributor disc is in line with the setting screw as shown. (For clockwise rotation use mark R, for counter clockwise, use mark L). With the armature in this position, the platinum contacts are just opening, and the metal insert of the distributor disc is in connection with carbon for No. 1 cylinder. The driving medium must now be fixed to the armature axle without disturbing the position of the latter, and the cables connected to the spark plugs.

of the plug and holding its end at a sufficient distance to allow a visible spark to leap from it to the plug core.

If this prove ineffective, the plug should be unscrewed and examined. Any visible fouling may then be removed by rubbing the insulation with fine sand paper until the bright surface of the porcelain is visible, taking care not to impair the surface.

If no fouling appear the plug may be laid upon the cylinder or frame so that its case only is in contact, and thus grounded, and on cranking the engine, the spark may be seen leaping between the points.

If a spark do not appear, it is probable that, with the ignition circuit in working order, there is some breakage or short circuit in the body of the plug. This, of course, necessitates its removal and the substitution of a new one. If a good spark appear, the search for trouble must be continued to other parts.

In a multi-cylinder engine a defective plug may sometimes be located by touch, that is, if its cylinder has been missing for some time, the metal of the plug will be perceptibly cooler than that of the other plugs.

Difficulty in Starting.—Sometimes an engine will start badly, but will run well after attaining a high speed. Among the various causes which contribute toward bringing about this condition may be mentioned:

1. *An obstruction* in the jet of the carburetter, causing trouble in starting, when removed by the suction allows the engine to speed up and run well at high speed;

2. *A too weak suction* in starting; this may be remedied by partially closing the air inlet while cranking, or giving the throttle more opening;

3. *Insufficient tension* of the auxiliary air valve spring.

4. *The spark* not sufficiently retarded.

Running Down.—When the engine starts well, runs for a while, then slows down and stops, there are several conditions to which it may ordinarily be attributed. Among these are:

1. Water or sediment in the carburetter;

2. Loose connections, breakdowns, or any other disarrangement of the ignition;

3. A weak or imperfectly recuperated battery, frequently the latter, that suddenly fails to supply current.

4. A leak in the water jacket that admits water to the combustion space;

5. Seizing or gripping of the piston in the cylinder on account of failure of the cooling system. This may result, in a water cooled cylinder, from:

a. Exhaustion of the water;
b. Stoppage in the pipes or pump;
c. Breakdown of the pump;
d. Failure of the oil supply.

In an air cooled cylinder seizing may result from:

a. Insufficient radiation surface;
b. Obstructed air circulation.

A precisely similar method may be followed in the search for a missing cylinder of a three or six cylinder engine.

A missing cylinder may also be found by the low temperature of its spark plug exhaust pipe, if the missing be long continued.

Misfiring During Operation.—Occasionally, the missing of one or more of the cylinders will be noticed during the operation of the engine. This trouble may be recognized by irregularity of motion, gradual slowing down, and, generally, by *after firing*, or explosions in the muffler. If the trouble cannot be located in one of the cylinders the inference holds: either that there is some general derangement of the ignition circuit, or that the fuel mixture is not right.

Running with Switch Off.—A peculiar condition in engine operation, sometimes encountered, is the running of the engine after the switch has been opened. It occasionally happens that the switch becomes defective, so that it does not break the circuit when in its "off" position. A most common cause for running with open switch is red hot plug points, also the heating to incandescence of some small particle in the cylinder, either loose or attached to the interior surfaces.

Pre-ignition.—An incandescent particle or overheated cylinder will cause an engine to pre-ignite.

Sometimes the rotor arm of the timer wears at the contact point leaving a path of metallic particles on the ring containing the stationary contacts, thus causing the current to flow to the stationary contact via this path and cause ignition to occur before the proper time.

Loss of Power Without Misfiring.—The chief cause for an engine to fail to deliver its full power is poor compression. A fuel mixture either too weak or too strong will reduce the power of the engine.

If the bearings be too tight there will be a loss of power due to the additional friction set up; bearings when too tight will heat. A touch of the hand will give indication of their condition.

Another source of loss of power is a defective clutch which slips and does not transmit all the power delivered by the engine.

Brake rods sometimes get out of adjustment, allowing the band to remain in contact with the drum, thus absorbing more or less power.

TROUBLE CHART FOR GASOLINE ENGINES

1. TROUBLES IN STARTING

(From Report by G. E. P. Smith, University of Arizona, Agricultural Experiment Station)

First, Turn Engine Slowly Through a Full Cycle and Note the Occurrence of Events

I. Ignition	1. (*a*) Battery	Test cells for amperage. (Reject a cell when below six amperes) Test storage batteries for voltage
	(*b*) Magneto	Poor contact of brushes Commutator smeared with oil Driven too slowly
	2. (For all engines)	Electric circuit not complete Poor contacts in wiring or in switch Wiring incorrect or short circuiting
	3. (*a*) Make-and-break Mechanism	Igniter mechanism out of time Igniter points roughened. (Use file or small hammer) Igniter points making poor contact on account of carbon deposit Igniter points short circuited by soot or oil, causing poor break Weak spring, causing slow break Insulation of stationary electrode broken
	(*b*) Jump-spark equipment	Induction coil vibrator stuck or out of adjustment Spark out of time Spark points short circuited by oil or carbon soot Spark points too far apart. Distance should be 1/16 inch or a little less Spark plug cracked and short-circuiting
II. Carbureter	1. Lack of gasoline	Tank empty Needle valve insufficiently open Inlet passages clogged Gasoline pump out of order Water accumulated in gasoline tank or in mixer
	2. Incorrect mixture of air and gasoline	Too rich Too weak (Prime it, but not too freely)
III. Engine	1. Cold Cylinder	
	2. Inlet or exhaust valve out of time	
	3. Very poor compression	Exhaust or intake valve does not close Piston rings much worn Cylinder gaskets blown out

2. TROUBLES AFTER STARTING

I. Misfiring	1. Ignition	Weak run-down batteries Poor connections Foul igniter points (or spark plugs)
	2. Carburetor	Lean mixture Gasoline tank nearly empty Fuel pump needs repacking Water in gasoline
	3. Motor	Cold cylinder Very poor compressions
II. Pounding	1. Spark too early	Should occur at from 10 to 20 degrees before dead center,
	2. Pre-ignition	Cylinder overheated Red hot carbon deposit in cylinder
	3. Looseness	Connecting rod bearings Main shaft bearings Fly wheel
	4. Hot Bearings	
	5. Tight Piston	No circulating water Cold water added suddenly when cylinder is hot may be the cause

TROUBLES AFTER STARTING—*Continued*

III. Back-firing	1. Late ignition 2. Weak mixture 3. Glowing points in cylinder	
	4. Inlet valve	Leaky Weak spring
IV. Explosions in Exhaust Pipe	1. Weak or irregular ignition 2. Weak mixture	
V. Over-heating	1. In cylinder	Late ignition Mixture too rich Insufficient lubricating oil Insufficient cooling water Impeded circulation of cooling water Poor compression
	2. In bearings	Lack of oil Screwed down too tightly
VI. Loss of Power	1. Ignition	Too late. (Advance it almost to where it causes pounding) Foul igniters (or spark plugs)
	2. Carburetor	Gasoline used up Passages partially clogged Needle and air valves must be set for running conditions Inlet valve does not lift sufficiently, because of wear
	3. Overheating	In cylinder In bearings
	4. Leaky compression	Valves do not close properly, springs gummy or weak, stems bent, valves or valve seats worn or pitted Piston rings gummy and sticking Piston rings and cylinder worn and out of circular shape Cylinder gaskets leaking
	5. Exhaust pipe 6. Engine overloaded	Gummy and partially clogged
VII. Smoke	1. Black smoke in exhaust	Mixture too rich
	2. Light bluish smoke in exhaust	Too much lubricating oil
	3. Escaping past the piston	Worn or stuck rings
VII. Irregular Speed	1. Irregular ignition	Weak batteries Loose electric contacts Worn make-and-break mechanism
	2. Irregular fuel supply	Fuel pump needs repacking
	3. Valves	Weak springs
	4. Governor	Stuck up with gummy oil and dirt, or with paint Out of adjustment Worn loose Needs lubrication

CHAPTER 47

OIL ENGINES

The constantly increasing cost of gasoline[†], has caused an insistent demand for engines that can operate on cheaper fuels, such as kerosene, heavy oils, etc. This demand has in turn caused manufacturers to give serious attention to the subject and many of the difficulties encountered have been overcome.

Oil.—In its broad meaning, oil is any greasy liquid and may be of animal, vegetable or mineral origin.

Mineral oil is any oil coming from the interior of the earth.

Crude oil, technically called petroleum, is the mother product or raw material just as it comes from the ground, from which the oil refinery produces all of its many products.

Oil refining is the process of separating the various grades of mineral oil contained in crude oil from the impurities therein

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[*] NOTE.—Everybody is wondering why gasoline is so high and increasing in price, and at the same time the answer is apparent to everyone. The countless number of automobiles that are decorating the highways and byways of this land and that seem to swarm like flies all over the northern and southern hemispheres, is constantly increasing every year. In 1911 there were about 500,000 automobiles in use in the United States; in 1912 about 800,000; in 1913 about 1,000,000, and in 1914 more than 1,200,000. In addition to automobiles, the number of motor boats, motor cycles and other gasoline burning engines is constantly increasing. Out of a gallon of petroleum that is pumped from the earth, only from 4 to 10% is available for gasoline, whereas the same gallon of petroleum will give from 44 to 50% of kerosene, the balance being made up of benzine, naphtha, distillate, also heavier oils. Consequently, practically half of the product of distillation is kerosene, while but 10% or less is gasoline. In other words, it is necessary to produce from four to ten gallons of kerosene to produce one of gasoline. There is not the market for kerosene that prevailed in other days, as householders do not use it as an illuminant in anything like the degree in years gone by, and outside of its use as an engine fuel the demand for kerosene is much less than ever before. This condition, together with the large amount of kerosene that is produced, accounts for its low price.

The Make up of Petroleum

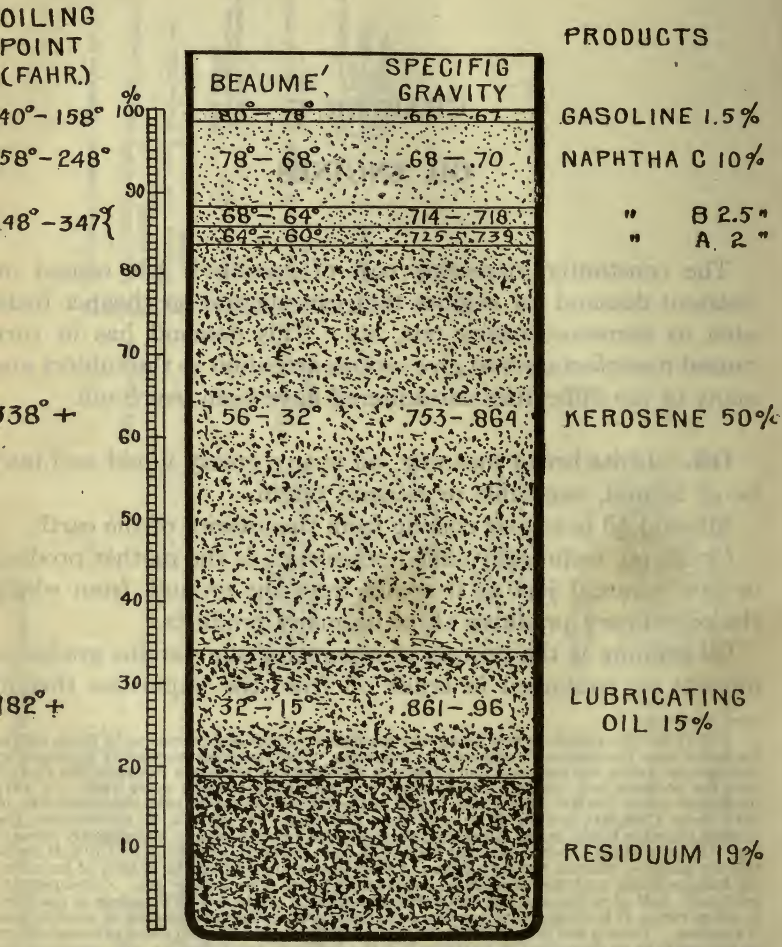


Fig. 2,840.—The various products obtained by the distillation of petroleum. In the process of distillation these products are separated according to their boiling points as indicated.

and classifying them according to the boiling point of each grade. In practice the crude stock is placed into a large vessel called a "still" and heated until all the oil, or as much of it as the refiner wants, distills out in the form of hot vapor or gas, which is afterwards condensed into a liquid state by cooling the vapor as it passes through coils surrounded by cold water. Since each commercial oil product has a different boiling or vaporizing temperature from any other, it is thus easy to break up crude and obtain in a clear refined state any of its constituents by taking out the lightest products first and then heating the residue to the boiling temperature of the next one desired.

Distillate in a broad sense is any product obtained by oil distillation. Strictly speaking the word distillate does not mean anything except that it is a clear product from an oil distillery. Those products lighter than kerosene are usually referred to, however, as distillates; those as heavy as kerosene and heavier are usually referred to as oils, although they are distillates as much as is gasoline. Distillate oil is the proper name.

Some clear refined distillates, arranged in the order of their boiling points, are:

Benzine.—A product lighter and more carefully refined than gasoline.

Gasoline.—A refinery product composed of a mixture of the various distillates in such proportions as to keep the density between 56° and 72° Baumè.

Petrol.—A foreign trade name for gasoline.

Benzol.—A light liquid product resembling gasoline which is obtained by fractional distillation of coal. In other words, a "by product" from gas works and bee hive coke ovens.

Motor Spirits.—A commercial name for a low grade gasoline recently put on the American market.

Naphtha.—Another trade name for cheap gasoline.

Distillate.—An engine fuel before given a commercial name.

Engine Distillate.—Trade name for kerosene too carelessly graded to be safe in lamps. (Gravity around 40° Baumè.)

Solar Oil.—Discolored, carelessly prepared kerosene.

Kerosene or Coal Oil.—A distillate heavier than gasoline and lighter than gas oil, bleached white and having a flash test high enough to pass the state oil inspector.

Gas Oil.—A distillate oil slightly heavier than kerosene and usually has an amber cast. One of the heaviest refined engine distillates.

Distillates heavier than gas oil are marketed for lubricating oil under various trade names. Those requiring the most heat to gasify and consequently coming off last are sold to lubricate hot surfaces such as engine cylinders and are known as high *fire test lubricating oils.*

Since the demand for lubricating oil is limited, only a small portion of the crude stock is worked down below the distilling points of the common engine distillates for which the refiner can get a ready market. In other words, he takes out of the crude those products his orders call for and dumps the residue into the fuel tank and sells it for fuel in the place of coal.

Some Cheap Unrefined Residual Oils.—Since these oils are scraps from the refinery and crude oils from the different fields vary in impurities and properties and different batches of crude receive different treatment to supply the refiners' demands, no two lots of fuel oil are usually similar in composition or density.

Fuel Oil.—Any unrefined oil that will do to burn in the place of coal.

Residium.—Another name for fuel oil.

Tops.—Another name for residium.

Stove Oil.—A trade name used in the West for fuel oil.

Star Oil.—A California trade name for fuel oil.

Tar Oil.—An oil obtained from crude coal tar in the coke and gas manufacturing industry.

Holder Oil.—Oil that condenses in the gas holder tanks. A by-product from city gas plants.

Gravity of Oils.—The standard instrument for the measurement of the density of oil is the Baume hydrometer. The gravity of oils should be taken at a temperature of 60° Fahr.

A rough rule for determining the gravity at any other temperature is: *For every* 10° *Fahr. above* 60°, *subtract one degree from the Baume reading, and for every* 10° *below* 60°, *add one degree.*

Example.—If the hydrometer indicate a gravity of 27.5° B. and the temperature of the oil for this reading be 75° Fahr., the temperature then is 15° above 60°, and the correct gravity is

$$27.5^\circ - \frac{15}{10} = 26^\circ \text{ B.}$$

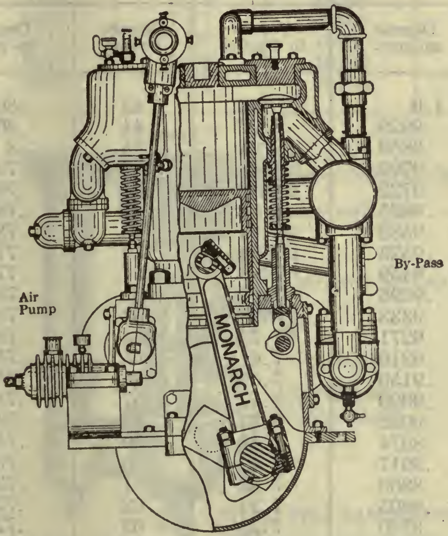


Fig. 2,841.—Sectional view of Monarch marine multiple fuel (gasoline, kerosene) engine showing by pass valve as described in fig. 2,842. Single fuel engine section would be the same except the inlet and exhaust manifold pipes would be different.

In case the hydrometer indicate a gravity of 24° B. at a temperature of 40° Fahr. Then the correct gravity will be

$$24 + \frac{60-40}{10} = 26^\circ \text{ B.}$$

Formula for reducing Baumè reading (liquids lighter than water) to specific gravity:

$$\text{specific gravity} = \frac{140}{130 + \text{degrees Baumè}}$$

Thus:

$$\text{gravity Baumè} = 26 \text{ at } 60° \text{ F.}$$

or

$$\text{specific gravity} = \frac{140}{130 + 26} = .897$$

Approximate Weights of Petroleum Oil

Degrees Baumè	Degrees sp. grav.	Pounds in 1 gallon	Degrees Baumè	Degrees sp. grav.	Pounds in 1 gallon
10	1.0	8.33	43	.8092	6.74
11	.9929	8.27	44	.8045	6.70
12	.9859	8.21	45	.8	6.66
13	.9790	8.16	46	.7954	6.63
14	.9722	8.1	47	.7909	6.59
15	.9655	8.04	48	.7865	6.55
16	.9589	7.99	49	.7821	6.52
17	.9523	7.93	50	.7777	6.48
18	.9459	7.88	51	.7734	6.44
19	.9395	7.83	52	.7692	6.41
20	.9333	7.78	53	.7650	6.37
21	.9271	7.72	54	.7608	6.34
22	.9210	7.67	55	.7567	6.30
23	.9150	7.62	56	.7526	6.27
24	.9090	7.57	57	.7486	6.24
25	.9032	7.53	58	.7446	6.20
26	.8974	7.48	59	.7407	6.17
27	.8917	7.43	60	.7368	6.14
28	.8860	7.38	61	.7329	6.11
29	.8805	7.34	62	.7290	6.07
30	.8750	7.29	63	.7253	6.04
31	.8695	7.24	64	.7316	6.01
32	.8641	7.20	65	.7179	5.98
33	.8588	7.15	66	.7142	5.95
34	.8536	7.11	67	.7106	5.92
35	.8484	7.07	68	.7070	5.89
36	.8433	7.03	69	.7035	5.86
37	.8383	6.98	70	.7000	5.83
38	.8333	6.94	75	.6829	5.69
39	.8384	6.90	80	.6666	5.55
40	.8235	6.86	85	.6511	5.42
41	.8187	6.82	90	.6363	5.30
42	.8139	6.78	95	.6222	5.18

Heat Value of Pure Oils of Different Gravities

Degrees (Baumé.)	Specific gravity	Weight per gallon	B.t.u. per pound	B.t.u. per gallon
10	1.0000	8.33	18280	152,272
12	.9859	8.21	18400	150,252
14	.9722	8.10	18520	148,033
16	.9589	7.99	18640	145,987
18	.9459	7.88	18760	143,927
20	.9333	7.77	18880	141,853
22	.9211	7.67	19000	139,956
24	.9091	7.57	19120	138,047
25	.9032	7.52	19180	137,088

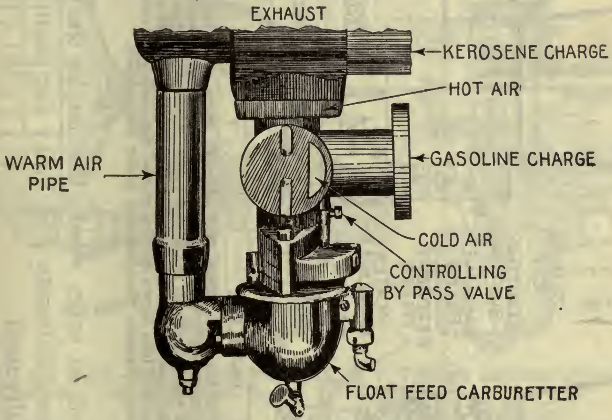
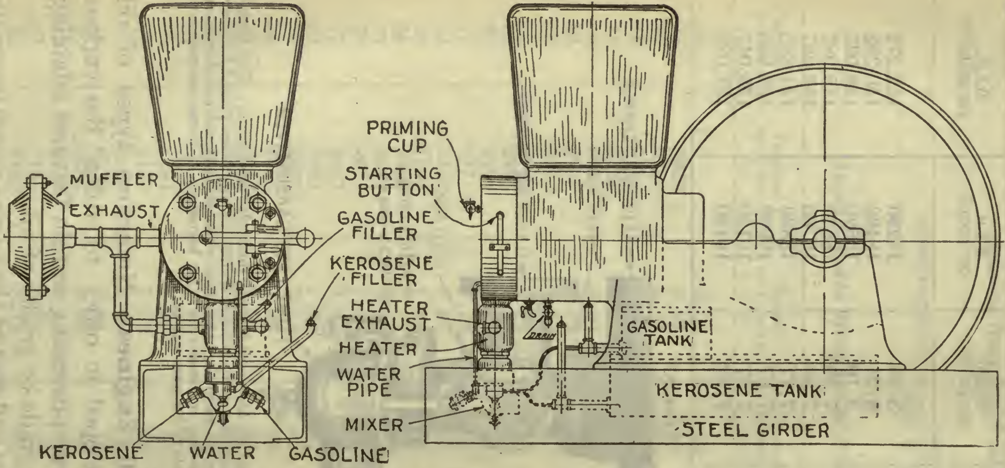


FIG. 2,842.—Monarch by pass as used on Monarch marine multiple fuel engine. This three-way valve is placed above the carburetter as shown and directs charge when placed horizontally, through cold air or lower pipe for gasoline, or when set vertically through annular space around hot exhaust pipe, super-heating and gasifying it for kerosene. Charge temperature may be regulated by adjusting angle of by pass valve.

Classification of Oil Engines.—Numerous types of oil engines have been developed in order to utilize the various grades of fuel oil, each type possessing certain features adapting it to the use of a certain grade or grades of oil.

In general the simplicity and cost of the engine varies inversely



FIGS. 2,843 and 2,844.—Woodpecker kerosene engine; end and side views. This engine is provided with a Lunkenheimer type mixing valve and an exhaust mixture heater between the valve and the engine as shown. The mixing valve has three needle valves for gasoline, kerosene and water as shown in the end view. After starting on gasoline, and heating the heating chamber the kerosene needle valve is opened and the gasoline closed. The water needle valve admits a spray of hot water for the jacket into the mixture which is evaporated into steam and by partial dissociation furnishes additional oxygen for combustion. It also helps to prevent excessive temperatures when working under heavy loads.

with the grade of the oil, in other words, the simpler the engine, the higher the grade of oil required; the more elaborate and bulky the engine, the lower the grade of oil which may be successfully used.

Oil engines may be classed:

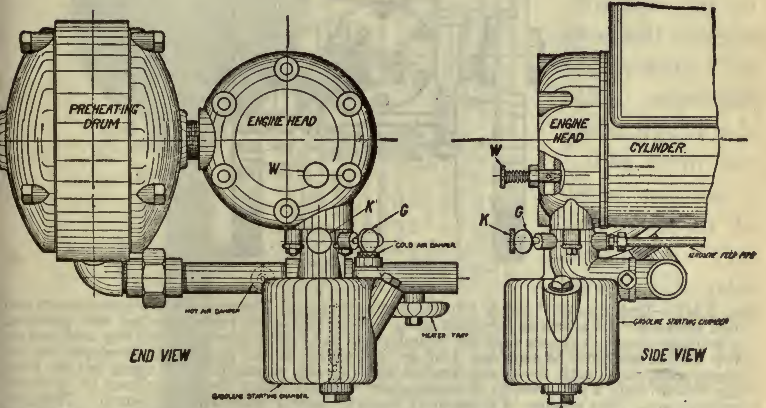
1. With respect to the density of the oil, as:

a. Light oil
  - kerosene;
  - distillate;

b. Heavy oil
  - crude oil,
  - fuel oil;
  - tar oil.

2. With respect to method of fuel supply, as:

a. Carburetter;
b. Pump;
c. Pump and compressed air.



FIGS. 2,845 and 2,846.—End and side views of R. & V. Triumph kerosene engine showing pre-heating drum, starting chamber, etc.

3. With respect to ignition, as by

a. Electric spark;
b. Heated metal (vaporizer);
c. Heat of compression.

4. With respect to the operating cycle:

a. Two (stroke) cycle;
b. Four (stroke) cycle;

c. Low compression { steam scavenging, pump injection;

d. Medium compression { combined pump and air injection or ***semi-Diesel.***

e. High compression: ***Diesel.****

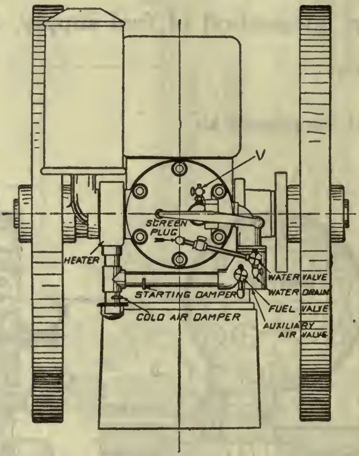


FIG. 2,847.—End view of Fuller & Johnson kerosene engine. The engine has a priming cup and fuel reservoir. ***In starting,*** the fuel pump is hooked up, the reservoir filled with gasoline, a priming charge is drawn into the cylinder and the engine started on battery and run on low tension magneto. ***Running.*** Hot air is supplied by the *heater* that envelopes the exhaust pipe, thus, with the engine started and running on gasoline and using hot air from the heater, only a *few minutes* of running are required before the engine may be operated on kerosene. The change to kerosene is made by simply unhooking the pump. This permits the pump to operate, and deliver the kerosene into the reservoir where it mixes with the gasoline, and the fuel is gradually changed to kerosene with *no throttle adjustments.* ***The mixer*** has primary and secondary air supply. On light loads all of the air enters through the regular air opening; on heavy loads (one-half to full load) the suction of the piston automatically opens the auxiliary air valve through which the additional required air is supplied. On heavy loads this cold air entering through the auxiliary valve tends to keep constant the high temperature of the cylinder. ***Water injection.*** To avoid excessive high temperatures in the cylinder when the engine is operating on heavy loads, water is injected along with the mixture. The water is supplied from the water jacket of the cylinder head, and is conducted by a short pipe (through a removable screen) to the water throttle. The water injection is automatically controlled by the auxiliary air valve, the water entering only when engine is under load (one-half to full load). The injection of water tends to lower the cylinder temperature, and to prevent formation of carbon. ***Throttling governor.*** In this method of governing the speed, the engine explodes every second revolution of the fly wheel, and the surplus heat generated by the successive explosions keeps the *cylinder* hotter, that is at a more economical temperature than the hit or miss method. The governor maintains constant speed by varying the strength of the explosions in proportion to the load; that is, the greater the load the wider the governor valve is opened, the greater the charge taken in, the higher the compression, and the stronger the explosion.

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*NOTE.—Owing to the importance of the Diesel engine it is treated in a separate chapter.

Kerosene Engines.—Kerosene as compared with gasoline, contains more carbon, and therefore, if proper conditions for combustion be provided, it should burn longer thus giving more power per gallon.

Kerosene is more difficult to vaporize than gasoline and unless proper methods have been employed in the design and construction, an engine will not give satisfactory results, especially under light loads.

The term kerosene engines is here used to denote the type whose operation *depends on preheating* the fuel for the mixture as distinguished from the heavy fuel types in which *the fuel is injected into the cylinder by means of a pump.*

Engines of the latter class however will operate on kerosene and are frequently used with that grade of oil.

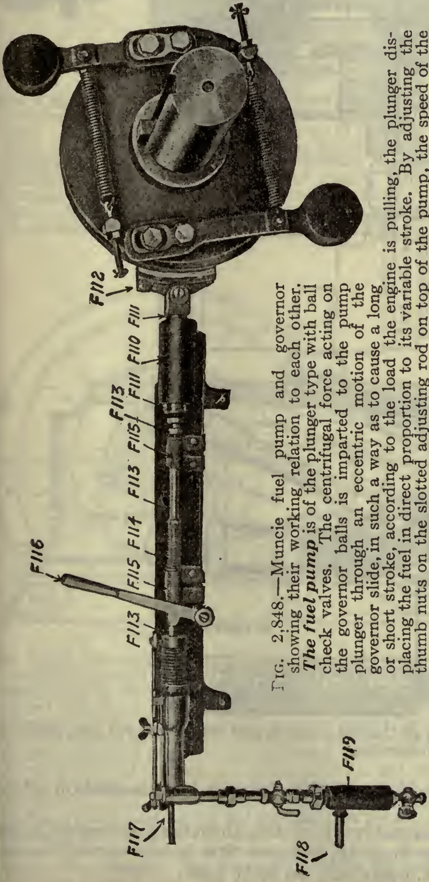
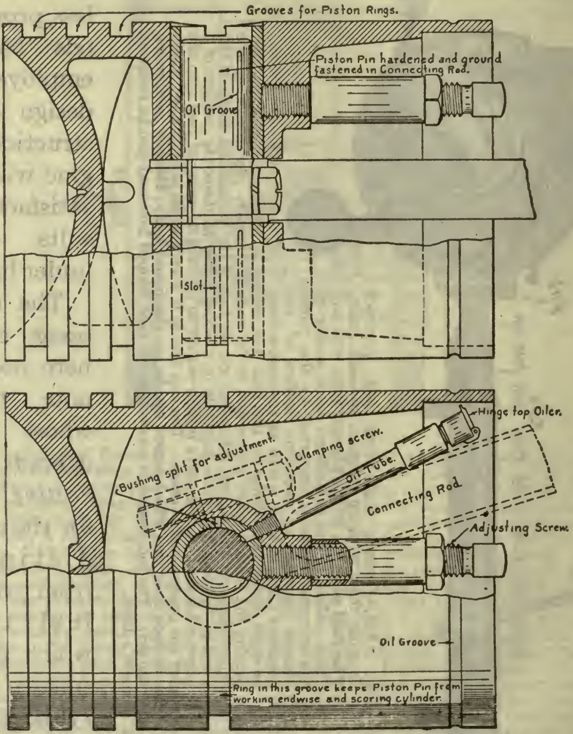


FIG. 2,848.—Muncie fuel pump and governor showing their working relation to each other. ***The fuel pump*** is of the plunger type with ball check valves. The centrifugal force acting on the governor balls is imparted to the pump plunger through an eccentric motion of the governor slide, in such a way as to cause a long or short stroke, according to the load the engine is pulling, the plunger displacing the fuel in direct proportion to its variable stroke. By adjusting the thumb nuts on the slotted adjusting rod on top of the pump, the speed of the engine can be varied at will of the operator while engine is in motion. ***The fuel nozzle*** consists of a steel body, fitted with a ground seated, spring valve inside, cut off to prevent dripping of fuel into the combustion chamber after the pump stroke ceases. The cap has in it a very small hole through which the pump forces the fuel at each injection, which action breaks up the liquid body, thereby performing the first agent of vaporization. The fuel nozzle is screwed into a pipe tapped hole and can be removed and replaced in a moment's time.

Figs. 2,845 and 2,846 show a kerosene engine designed to operate on a pre-heated mixture. As will be seen there is a carburetter or mixer and a heater.

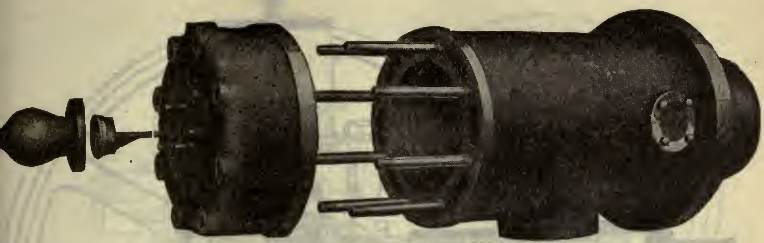
The mixer is provided with two fuel nozzles and a water nozzle. The engine is started on gasoline and run on kerosene after the parts have become sufficiently heated.

The water nozzle is used to introduce water into the mixture to prevent excessive temperature on heavy loads and to furnish by dissociation additional oxygen for combustion.


Grooves for Piston Rings.
Piston Pin hardened and ground fastened in Connecting Rod.
Oil Groove
Slot
Hinge top Oiler.
Clamping screw.
Bushing split for adjustment.
Oil Tube.
Connecting Rod
Adjusting Screw.
Oil Groove
Ring in this groove keeps Piston Pin from working endwise and scoring cylinder.

FIGS. 2,849 and 2,850.—Sectional views of Domestic piston and connecting rod end, showing unit pin adjustment, oiler, ring and oil grooves.

Kerosene engines should preferably be of the throttling governor type in order to insure adequate pre-heating, because this is not so well maintained in the hit and miss type, especially on light load.



Figs. 2,851 to 2,854.—Muncie igniter head, igniter spoon, and cylinder disassembled showing construction. The igniter spoon is hollow, having a flat surface on forward end and flange on rear. It is passed through the opening in cylinder head. Over this is placed the igniter head, which has a larger flange that is secured to the cylinder head by means of studs, thus holding the spoon firmly in place. Between the joint to make it tight and hold the compression is placed a copper asbestos gasket. In this manner there is made a complete passage from the cylinder, the combustion chamber, through the spoon and into the igniter head. Over the igniter head is placed a cast iron hood, lined with asbestos to hold the heat in the igniter head.

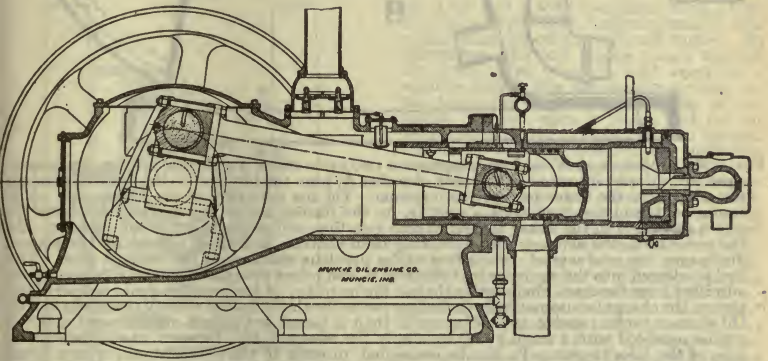


Fig. 2,855.—Muncie low compression two cycle steam scavenger engine. ***The steam*** for the explosive mixture is taken into the combustion chamber in the form of water spray at the same time as the air. It is admitted by means of an adjustable valve on top of cylinder, called the bleeder valve. This water entering the combustion chamber and changing to steam while the exhaust port is yet open assists the incoming air to more thoroughly sweep out the burned gases. The fuel is not put into the combustion chamber until both ports have been closed and the air and steam have been partly compressed at which time it is forcibly injected into the hot compressed air and steam, onto red hot cast iron vaporizing surfaces inside of the combustion chamber, which surfaces are kept incandescent by the explosions. The heat caused by compressing the air, plus the heat absorbed from the hot surfaces inside, assisted by red hot cast iron for any liquid globules to fall onto, is sufficient to gasify any oil as soon as injected into the combustion chamber. The red-hot surfaces, in addition to assisting in making gas out of the oil for the mixture, also automatically ignite the charge. The governor controls the speed by regulating the length of the fuel injection pump stroke, thus varying the richness of the mixture.

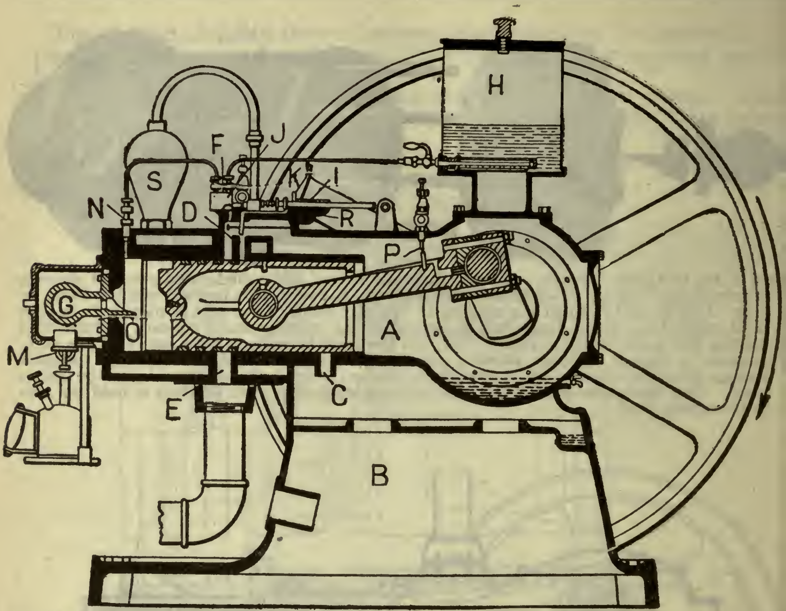
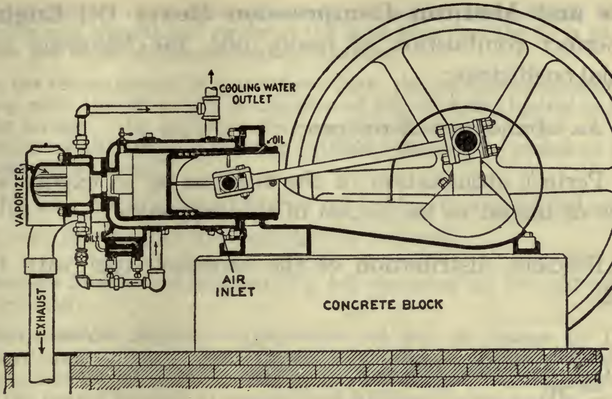
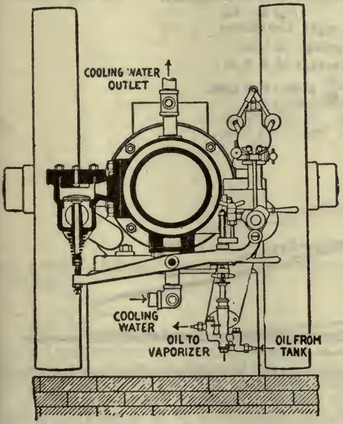


FIG. 2,856.—Meitz and Weiss low compression two cycle steam scavenging engine. ***In operation,*** air is drawn into the closed crank chamber A, from the interior of the base B, through the port C, in the lower part of the cylinder. On the outstroke of the piston, this air is compressed, and the opening of a port D, by the piston, allows the air, together with the steam generated in the water jacket, to pass into the combustion space of the cylinder. At the same time, the exhaust port E, having been overrun, and thus opened by the piston, discharges the products of combustion of the previous charge into the exhaust pipe. The fuel is injected into the cylinder by the pump F, and mixes with the air and steam previously admitted from the crank chamber, so that on the instroke, or compression stroke proper of the piston, the charge is automatically ignited by contact with the heated walls of the igniter ball G. Oil storage tanks capable of holding more than a barrel are placed underground and the engine provided with a small pump to supply the reservoir H. In most cases, the suction pipe of the injector pump F, can be connected directly to the storage tank, or to a small tank placed on the floor or on the engine. The oil pump is of the single acting plunger type, the suction and pressure checks being screwed into the pump body. A small lever serves to operate it by hand in starting the engine, and to draw the oil in case the pipes are not completely filled. During continuous running, it is operated by a rocker arm connected to the eccentric through the plunger guide I. The spring around the plunger L, forces it back to the short spring stop of the guide. ***To stop the engine,*** the pump lever is pulled back while a pressure is exerted on the small projecting pin K, at the side to lock the plunger guide out of action. ***Ignition*** is effected by means of a hollow cast iron ball G, located in the projection attached to the cylinder head. A charge is formed at every revolution of the crank shaft and compressed by the piston into the compression space of the cylinder and the interior of the igniter ball, where it is promptly ignited when the piston reaches the dead point. ***Before starting,*** the igniter ball is heated for a few minutes by a small oil burner, M. The oil jet from the injection nozzle, N, strikes the projection O, extending from the igniter ball and is sprayed, vaporized and mixed with the air and steam in the compression space. The igniter ball is maintained at a dull red heat by the heat of the explosions. The cylinder is water jacketed.





FIGS. 2,857 and 2,858.—Sectional views of De La Vergne (type HA), low compression, four cycle, pump injection, heavy oil engine. ***In operation,*** the fuel charge enters through the inlet valve which is held open by a suitable cam during the suction stroke. At the beginning of the return stroke of the piston, the air valve is closed and the air thus confined in the cylinder is forced through the narrow passage shown in fig. 2,857 into the vaporizer and compressed to about 75 lbs. pressure, and ignited by the combined heat of compression and that of the hot walls of the vaporizer. The vaporizer consists of two parts, a jacket and a cap. The jacket is bolted to the cylinder and connects with it through the narrow passage shown in fig. 2,857. The cap consists of a massive gun iron thimble, heavily ribbed on the inside to increase its surface and is bolted to the jacket. The cap is unjacketed and is maintained at a high temperature by the combustion of the successive charges of fuel by which the engine is driven. The vaporizer must be heated to a dull red heat by a blast lamp before the engine is started, but the use of the blast lamp may be discontinued as soon as the engine is running. The power and exhaust follow in regular sequence. The exhaust valve is operated by a lever driven from the cam shaft, located under the cylinder side by side with the lever which operates the air inlet valve. The exhaust gear is such that when starting, the compression may be relieved by shifting the exhaust cam on the cam shaft. The governor controls the speed by regulating the amount of oil fed to the vaporizer in proportion to the work to be done. This engine will not only run successfully on kerosene or any of the various crude oils produced in the Eastern and Middle States, but almost all of the so-called distillates, and residual fuel oils, 24° Baumè or lighter, may be employed.

Low and Medium Compression Heavy Oil Engines.—For perfect combustion of heavy oils, the following are the essential conditions:

1. An abundance of oxygen;

2. Perfect atomization of the fuel, so as to expose a large surface of the oil to the action of the oxygen;

3. Efficient distribution of the atomized fuel with the air charge;

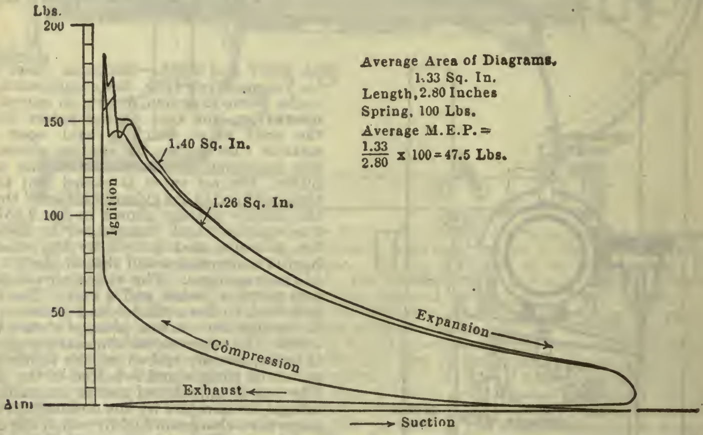


FIG. 2,859.—Indicator diagram of the De La Vergne (type HA) low compression four cycle pump injection heavy oil engine.

4. Highly heated vaporizer chamber.

These conditions may be produced in either two or four cycle operation.

Fig. 2,856 is an example of low compression two cycle heavy oil engine and shows one arrangement of the vaporizing chamber.

In operation oil is sprayed into the vaporizer immediately after the completion of the exhaust stroke at the same time that the oil is vaporized by the hot walls of the combustion chamber.

On the return stroke, the air is forced into the combustion chamber, and, mixing with the oil vapor, is compressed with it, being ignited at the end of the stroke. The vaporizer here mentioned may be called an *internal* vaporizer as distinguished from an *external* vaporizer or preheater as used on kerosene engines.*

In the four cycle type of heavy oil engine the essential requirements just mentioned are fulfilled as follows:

Suction Stroke.—Admission of a full charge of air into the vaporizer and cylinder.

Compression Stroke.—Compression of the air charge to 150 lbs. pressure (more or less). Near the end of the stroke, the oil is injected into the vaporizer in the form of spray by the fuel pump. Coming into contact with the heated air, rich in oxygen, and striking the hot surface, vaporizes and ignites.

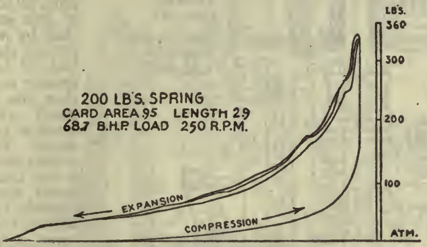


FIG. 2,860.—Indicator diagram from Venn-Severin type K medium compression two cycle oil engine.

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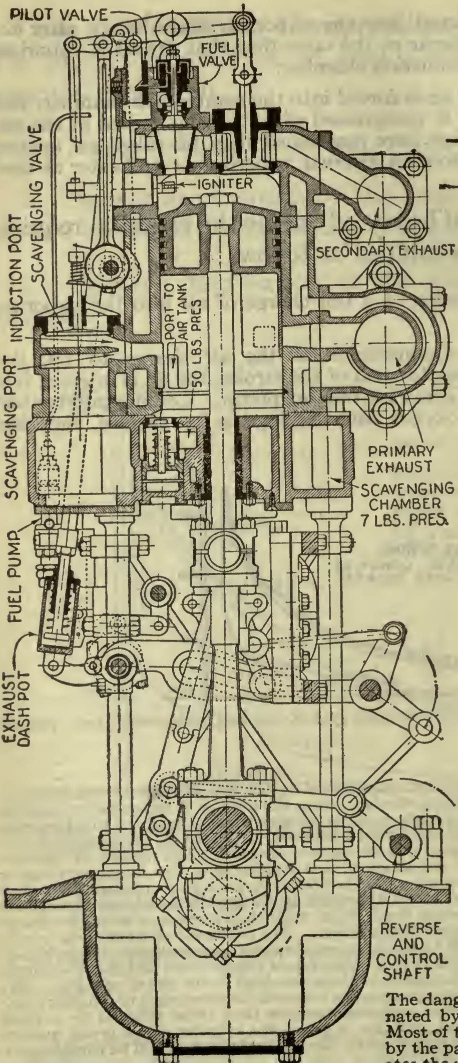
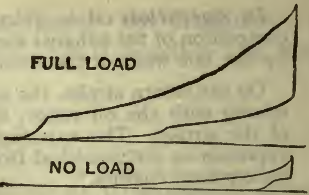
*NOTE.—One of the main difficulties encountered in the use of external vaporizers or mixture pre-heaters in oil engines is due to the fact, that, the process of vaporization requires the raising of the temperature of the oil vapor and the air with which it is mixed to a temperature ranging from 250° to 300° Fahr., at atmospheric pressure. As a result, the specific gravity of the charge is greatly reduced, so that a given volume contains a less number of thermal units than when it is at atmospheric temperature, and furthermore, the consequent higher initial temperature results in a higher compression temperature for a given pressure than that attained in the case of a gas engine. The total result is a higher temperature throughout the cycle and a greater loss of heat through the cylinder walls than is the case when the charge is admitted to the cylinder at atmospheric temperature. The combined effect of these conditions is the imposition of an early limit to the compression ratio, on account of the increased liability to premature ignition due to the higher compression temperature and the lower ignition temperature of the oil vapor, as compared to those involved in the use of coal gas. Engines which do not use external vaporizers possess the inherent disadvantages of a higher compression pressure accompanied by a lower explosion pressure than those attained by engines of the vaporizer type. This indicates, that for a given size cylinder and a given quality of fuel, the mean effective pressure and consequently the power developed will be considerably lower in the former than in the latter.





Figs. 2,861 to 2,863.—Raabe variable compound compression, scavenging, reversible, marine oil engine. The easy working cycle and special valve gear permit a wide speed range, thus adapting it to lighters, tugs and other heavy working boats requiring quick control in manœuvring around docks, etc. The indicator cards, figs. 2,862 and 2,863, show full and no load operation. There is one impulse per revolution as in an ordinary two stroke cycle engine, although the actual cycle of this engine places it in a class by itself. The engine will run on the heaviest grades of fuel oil including tar oil with greater economy (as claimed) than the Diesel and permit a speed variation from rated speed to less than 5% of rated speed. For heavy duty the weight and size of these engines is less than 50% and for high speed work such as yachts torpedo boats, etc., less than 25% of that of Diesel engines of equal capacity. ***In operation,*** the oil is fed to the fuel valve by the fuel pump and fuel valve transforming the oil into an almost perfect gas during the charging period of the combustion chamber where it is mixed with the air previously admitted during scavenging. This explosive mixture is then ignited by electric ignition as in an ordinary gas engine and perfect combustion is obtained, due to the perfect transformation of of the oil from the liquid to the gaseous state. ***In starting,*** the fuel atomizer is electrically preheated; afterwards this heat is maintained by combustion. The danger of premature explosions is eliminated by the compound compression system. Most of the scavenging air enters the cylinder by the patented induction system which eliminates the use of large scavenging pumps.

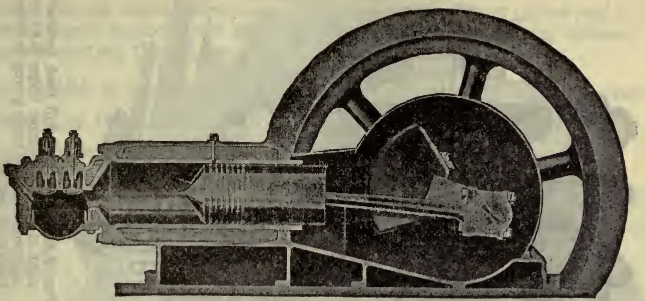


FIG. 2,864.—De La Vergne (type DH) medium compression four cycle pump injection heavy oil engine.

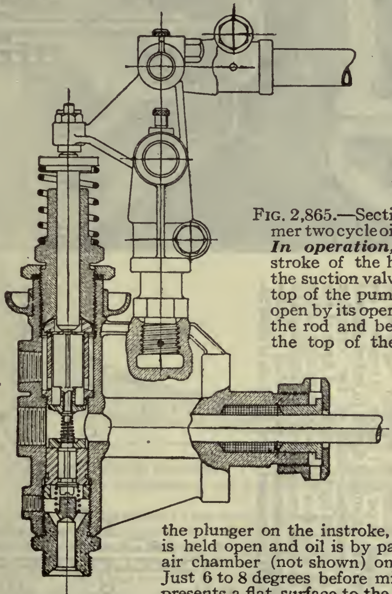


FIG. 2,865.—Sectional view of Bessemer two cycle oil engine fuel pump. ***In operation,*** during the outstroke of the horizontal plunger the suction valve opening into the top of the pump chamber is held open by its operating cam through the rod and bell crank shown at the top of the pump, the cam itself being mounted on the crank shaft. During this time an excess charge of oil is drawn into the cylinder. As the pump eccentric reaches its dead center and starts the plunger on the instroke, the suction valve is held open and oil is by passed back into the air chamber (not shown) on the suction pipe. Just 6 to 8 degrees before mid-stroke, the cam presents a flat surface to the roller which takes off the motion, allowing a very rapid closure of the suction valve, while the continued advance of the plunger forces the lower or discharge valve open, delivering oil to the combustion chamber for a period of 12 to 16 degrees, which is terminated by the re-opening of the upper valve by the cam motion again allowing the oil to by pass. By the use of a cam controlled injection period, the timing of the injection is determined independent of the governor.

Power Stroke.—The resulting pressure of combustion drives the piston forward, transmitting energy to the fly wheel.

Exhaust Stroke.—The burnt gases are expelled from the cylinder.

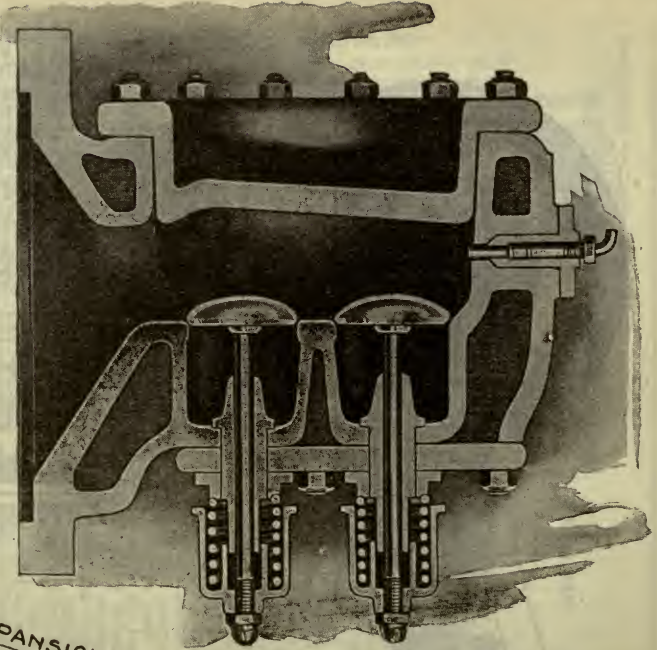


FIG. 2,867.—Cross section through vaporizer of De La Vergne (type DH) heavy oil engine. ***In operation*** the atomized fuel is injected against the hot (unjacketed) walls of the vaporizer where it is ignited by the combined heat of the valves and that due to the compression of the charge. In this arrangement pre-ignition and back firing are impossible as no fuel is introduced until the moment when ignition is desired.

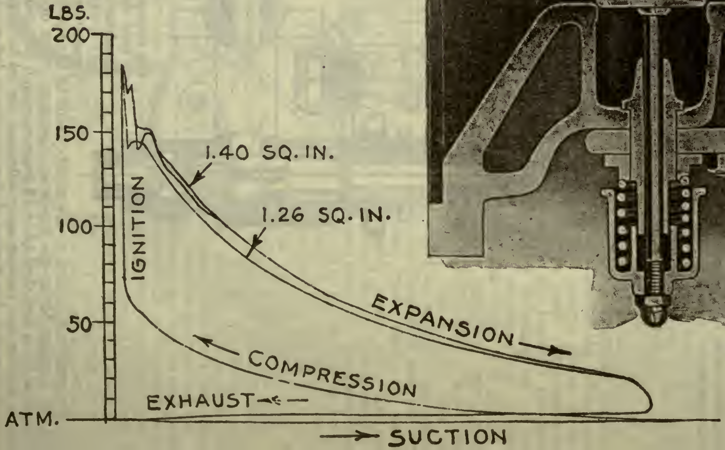


FIG. 2,866.—Typical full load indicator card of De La Vergne (type HA) low compression four cycle pump injection heavy oil engine.

Usually the air valve opens a few degrees before exhaust closes and a charge of fresh air is sucked into the vaporizer by the out rushing exhaust gases thus completing the scavenging.

Fig. 2,864 is an example of heavy oil engine of the medium compression type, and fig. 2,860 a typical indicator card of a medium compression engine.

Semi-Diesel Oil Engines.—This classification is applied to engines of the medium compression heavy oil type in which

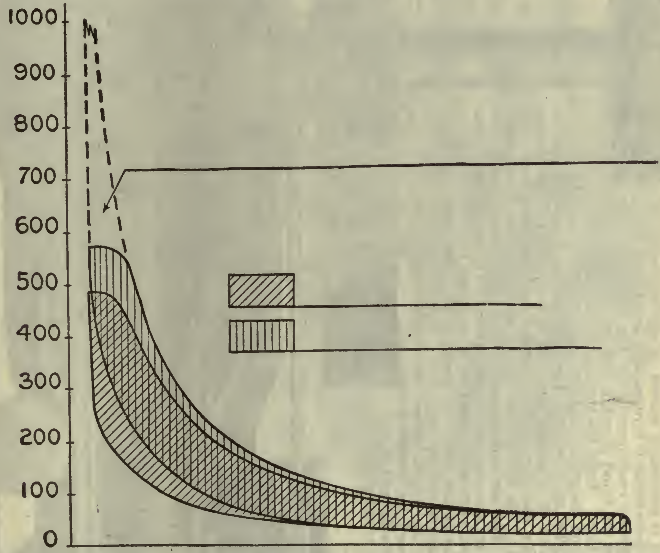


FIG. 2,868.—Comparison of indicator cards showing: 1, high compression cycle; 2, De La Vergne compression cycle, and 3, high compression cycle with faulty spray valve adjustment.

NOTE.—***How to start a low compression two cycle oil engine.*** Previous to starting the engine, the igniter head is heated a few minutes by means of a gasoline blow torch, oil burner, gas burner or any other convenient method. When the igniter head is sufficiently heated and with the piston in proper position (ports closed) simply give the starting lever on the fuel pump a couple of sharp strokes to inject the initial charge of fuel into the combustion chamber. Then (in starting by hand) reverse flywheel quickly, against compression, which compresses and ignites the charge, and the engine is under way, the fuel injection and ignition occurring thereafter automatically. In starting by air set the flywheel so that the crank is just above back center, upon which a quick opening and closing of the lever handle throttle admits sufficient volume of air which under expansion quickly drives piston forward, setting flywheel in motion. After starting the torch or burner is extinguished.

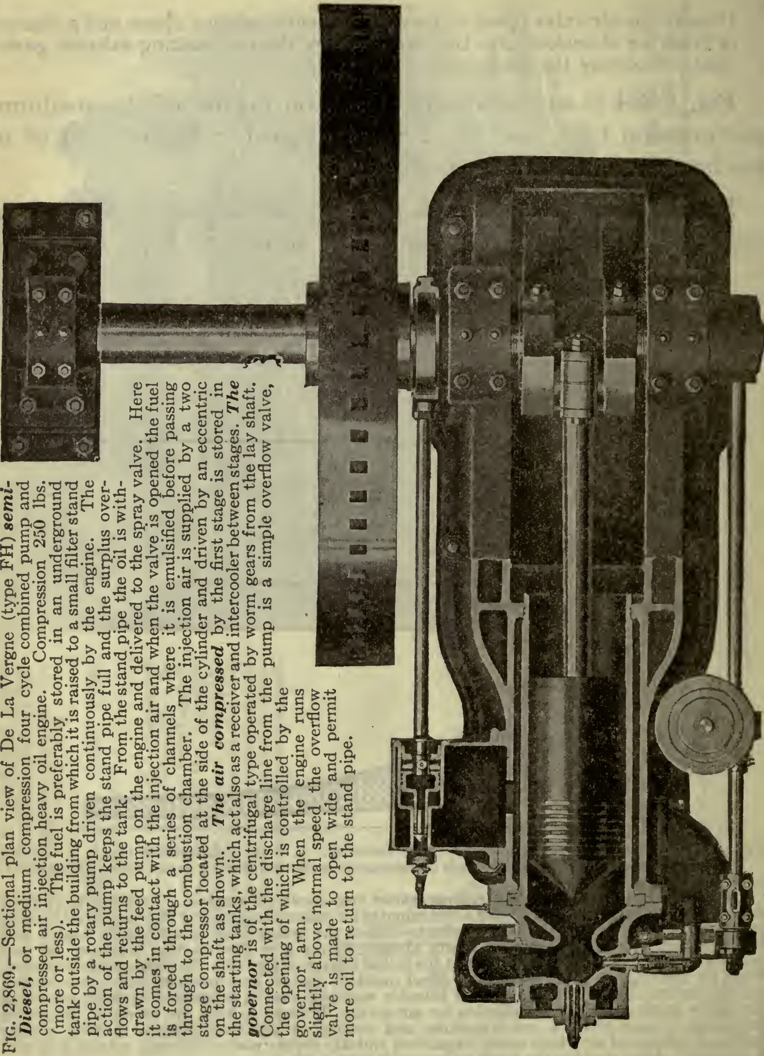


FIG. 2,869.—Sectional plan view of De La Vergne (type FH) ***semi-Diesel,*** or medium compression four cycle combined pump and compressed air injection heavy oil engine. Compression 250 lbs. (more or less). The fuel is preferably stored in an underground tank outside the building from which it is raised to a small filter stand pipe by a rotary pump driven continuously by the engine. The action of the pump keeps the stand pipe full and the surplus overflows and returns to the tank. From the stand pipe the oil is withdrawn by the feed pump on the engine and delivered to the spray valve. Here it comes in contact with the injection air and when the valve is opened the fuel is forced through a series of chambers where it is emulsified before passing through to the combustion chamber. The injection air is supplied by a two stage compressor located at the side of the cylinder and driven by an eccentric on the shaft as shown. ***The air compressed*** by the first stage is stored in the starting tanks, which act also as a receiver and intercooler between stages. ***The governor*** is of the centrifugal type operated by worm gears from the lay shaft. Connected with the discharge line from the pump is a simple overflow valve, the opening of which is controlled by the governor arm. When the engine runs slightly above normal speed the overflow valve is made to open wide and permit more oil to return to the stand pipe.

the fuel is injected by the combined pump and compressed air method. Engines of this class like those of the preceding type depend on the combined heat of compression and that of the vaporizer for ignition.

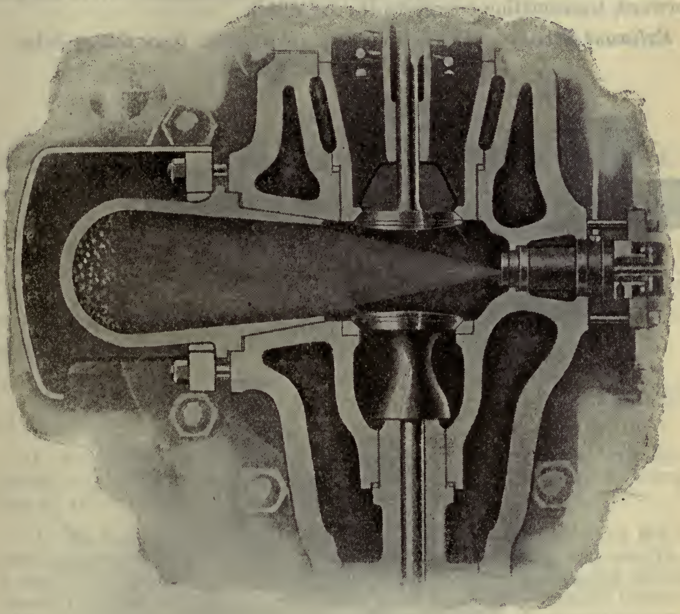


FIG. 2,870.—Cross section through cylinder of De La Vergne (type FH) ***semi-Diesel engine*** showing combustion chamber and valves. ***In starting,*** the combustion chamber is heated by a blast lamp for from 7 to 15 minutes and discontinued when the engine starts. The combustion of the fuel then maintains the chamber at the proper temperature. The engine is started automatically by admission of the air from the air tanks furnished for that purpose. ***To start,*** heat combustion chamber with the blast lamp, place crank in starting position and open the air cock. A cam on the lay shaft then successively opens and closes the starting valve at the proper times. When combustion has commenced and the engine begins to gather speed, the air cock should be closed. ***To stop,*** cut off the supply of fuel.

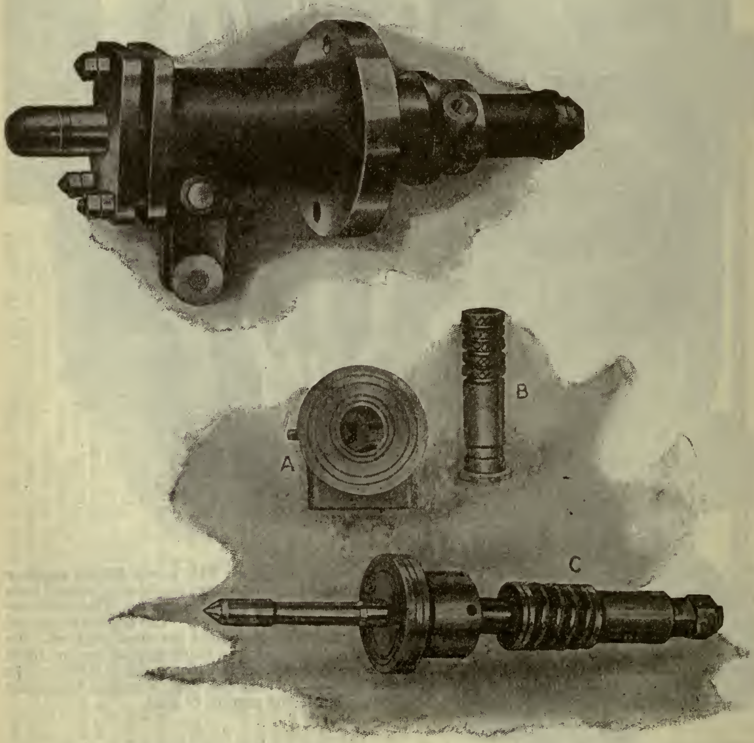
The operating cycle requires four strokes as follows:

Suction Stroke.—Admission of a full charge of air into the cylinder.

Compression Stroke.—Compression of the air charge into the combustion space to 250 lbs. (more or less). At or near the end of the compression stroke, the fuel oil is delivered to the spray valve by the fuel pump and blown into the cylinder by air furnished by a compressor at a pressure above that due to compression in the engine cylinder. The charge thus injected into the cylinder being ignited by the heat of compression.

Power Stroke.—The resulting pressure of combustion drives the piston forward, transmitting energy to the fly wheel.

Exhaust Stroke.—The burnt gases are expelled from the cylinder.



Figs. 2,871 to 2,874.—De La Vergne (type F. H.) ***semi-Diesel engine,*** pulverizer or spray valve. Fig. 2,871, assembly in casing; fiig. 2,872, stuffing box; fig 2,873, atomizer; fig. 2,874, spray valve assembled.

CHAPTER 48

DIESEL ENGINES

Rudolph Diesel, a German engineer, aiming to increase the efficiency of internal combustion engines, found from purely theoretical consideration that the highest efficiency would be realized by carrying out in practice the ideal Carnot cycle.*

To approximate as nearly as possible to this cycle Diesel devised a cycle, the unique features of which he describes in his treatise "The Rational Heat Motor," as follows:

> "The characteristics of the working cycle, as deduced from one theory of combustion are:
>
> "1. Production of the highest temperature of the cycle, not by and during combustion, but before and independently of it, solely by compression of air, or of a mixture of air and gas.
>
> "2. Gradual introduction of the pulverized combustible into the compressed and heated air during part of the stroke of the piston, and in such a way that combustion produces no increase in temperature of the gases. Hence the combustion curve obtained is as nearly as possible isothermal. After ignition, combustion is not left to itself, but is regulated by external mechanism throughout its course, and the right proportions are thus established between the pressures, volumes and temperatures of the gases.
>
> "3. Choice of the right weight of air in proportion to the heat value of the combustible, the temperature of compression (and of combustion) being

---

*NOTE.—In the ***Carnot cycle,*** a working fluid at a given temperature and volume expands while doing work and receiving heat sufficient to maintain it at constant temperature. When the addition of heat ceases the expansion continues adiabatically, the temperature falling to the lowest point of the cycle. During the return, heat is withdrawn in proportion to the reduction in volume, and the curve is again isothermal. The compression is adiabatic, and in the true Carnot cycle the gas is returned to its original volume and temperature.

previously so determined that the working of the engine, lubrication, etc., can be carried out without cooling the cylinder.

"We need scarcely say that, in practice, these conditions can hardly be realized. In working engines the corners of the indicator diagrams are more or less rounded, the curve of combustion is not perfectly isothermal and under given conditions a certain increase in temperature or pressure will be caused by combustion. This rise, however, is slight in comparison, with that previously produced in the temperature and pressure of the gases by compression. Nor do these differences affect the principle of the cycle."

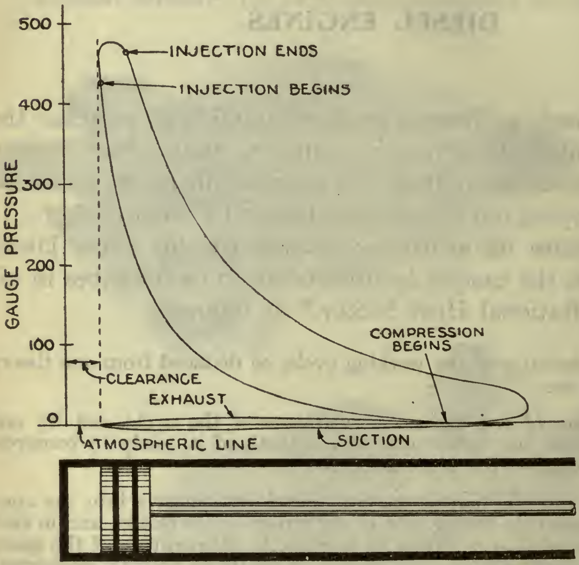


FIG. 2,875.—Typical indicator card of Diesel four cycle engine. It will be noted from the diagram that the pressure range and mean effective pressure is much greater than in other engines which accounts for the massive construction.

The Diesel cycle may be completed in either two or four strokes, the latter being the prevailing practice. Briefly the four stroke Diesel cycle is as follows:

Suction Stroke.—Admission of air into the cylinder.

Compression Stroke.—Compression of the charge of air to about 500 lbs. pressure which causes its temperature to rise to about 1,000° Fahr.

As this pressure is reached gradually it does not cause a shock to the engine, such as an explosion to the same pressure would give.

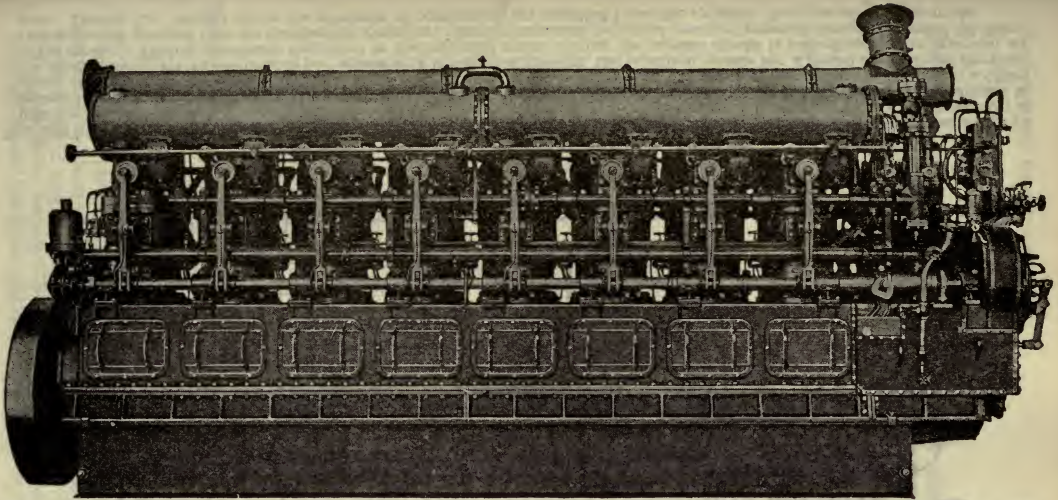


FIG. 2,876.—Nelseco four cycle marine Diesel engine, direct reversing type (exhaust side), built in 360, 480, 600, 800, and 1,000 horse power sizes.

Power Stroke.—At the beginning of the stroke, oil previously delivered to the injection valve is blown into the cylinder in the form of fine spray by a small quantity of air compressed by a special compressor to 700 or more lbs. pressure. The oil spray meeting the highly heated air in the cylinder ignites and burns, combustion continuing so long as the fuel is being injected, usually for about one tenth of the power stroke.

Usually the heat generated by the combustion is not sufficient to prevent the pressure in the cylinder falling while admission is taking place, so that the admission line on the indicator card falls below the constant pressure line as seen in the indicator card, fig. 2,875.

Exhaust Stroke.—Expulsion of the products of combustion from the cylinder, this completing the cycle.

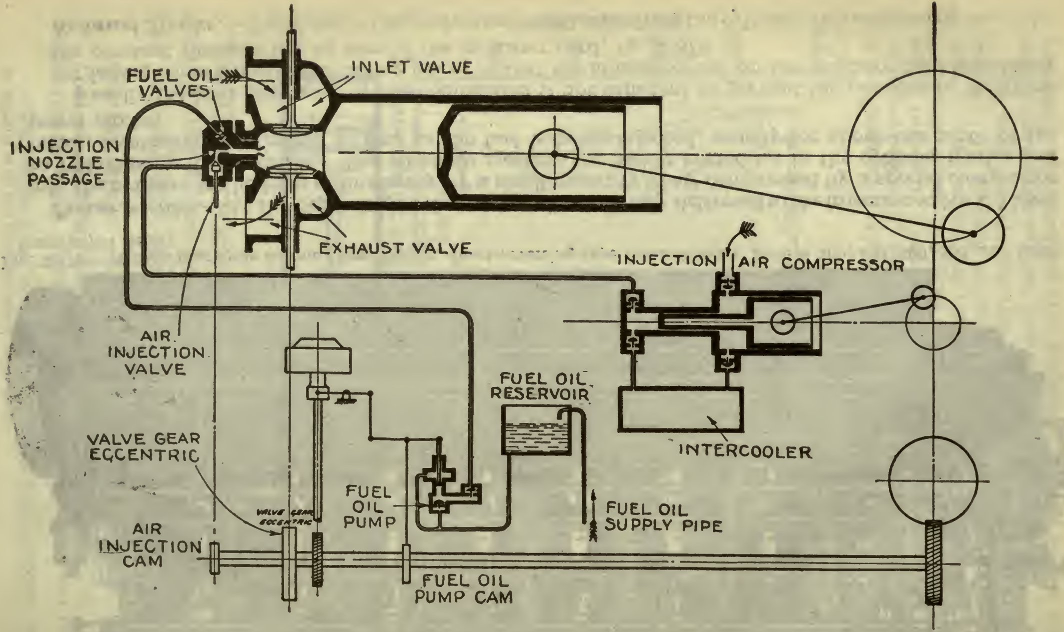


FIG. 2,877.—Schematic arrangement of Allis-Chalmers Diesel engine. ***In operation,*** during the *suction stroke* pure air is drawn into the cylinder through the main inlet valve and a proper charge of fuel oil (determined by the position of the governor) is pumped without pressure into the injection nozzle passage. On the *compression stroke*, the air in the cylinder is compressed to about 500 lbs. pressure, which gives it a temperature in excess of 1,000° Fahr. At the *beginning* of the *power stroke*, air from the compressor at 700 to 900 pounds per square inch is admitted through the injection nozzle for a period of about ten per cent. of the stroke. The oil previously deposited in the injection nozzle during the suction stroke is picked up by the injection air and gradually sprayed into the combustion chamber, where combustion occurs at constant pressure during the period of injection. During the *exhaust stroke* the products of combustion are expelled from the cylinder, thus completing the cycle.

3. Sprayer or pulverizer;
4. Air compressor;
5. Valve gear and connections.

These essential parts are shown in the schematic diagram, fig. 2,877, showing arrangement of the Allis-Chalmers Diesel engine.

Diesel engines are manufactured in both vertical and horizontal types.

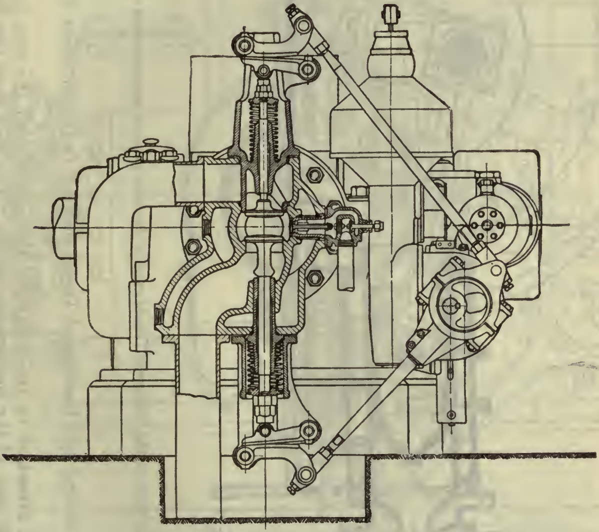


FIG. 2,878.—Cross section through valve gear of Allis-Chalmers Diesel engine.

The advantages of the horizontal engine, as enumerated by a concern with an established reputation as builders of vertical engines are: Lower price; easy survey and convenient attendance; better distribution of stresses on the crank shaft and main bearings; better lubrication of bearings; better lubrication of piston; easier cleaning and repair work, especially to pistons which can be taken out without removing the cylinder head and the valve gear; reduced height of engine room, as all pipes are laid in trenches under the floor and do not obstruct the room.

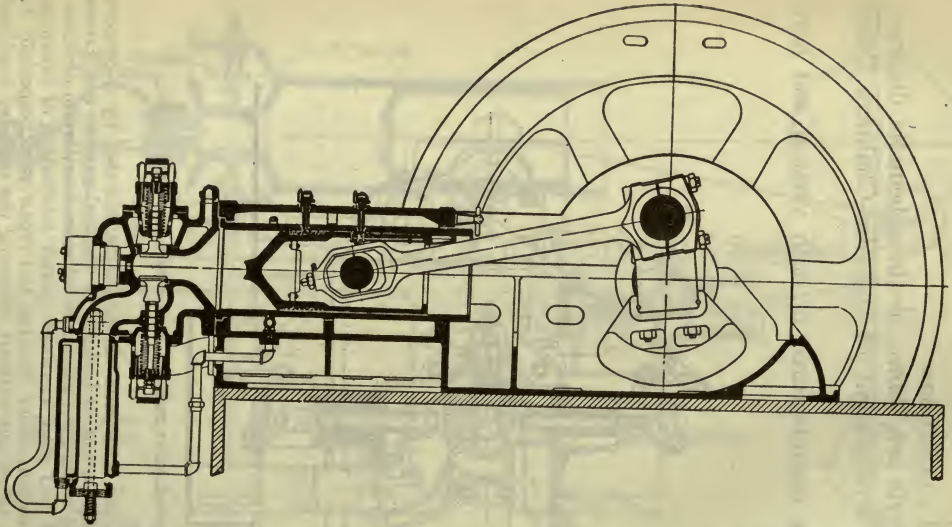


FIG. 2,879.—Longitudinal section of the National Diesel engine showing general arrangement of parts.

In a comparison of vertical and horizontal Diesel engines the detail most discussed is the piston.

Vertical builders call attention to the weight of the piston, and the side pressure thereby produced on the cylinder bore, in the case of the horizontal engine. The fact that the first Diesel engines were built vertical, and that excessive height was undesirable, probably led to the use of trunk pistons. For the same reason a connecting rod of a length but slightly more than five cranks has been used in vertical engines.

Regardless of the merits of the two types the fact remains that the majority of Diesel engines are of the vertical type.

The Cylinder.—The usual jacketed cylinder is employed in the Diesel engine, though it is somewhat longer than the ordinary cylinder to accommodate the extended piston. The cylinder head, which is also water jacketed, contains the inlet valve, exhaust valve, air starting valve, and fuel sprayer. Figs. 2,879 and 2,880, which are respectively longitudinal and transverse sectional views

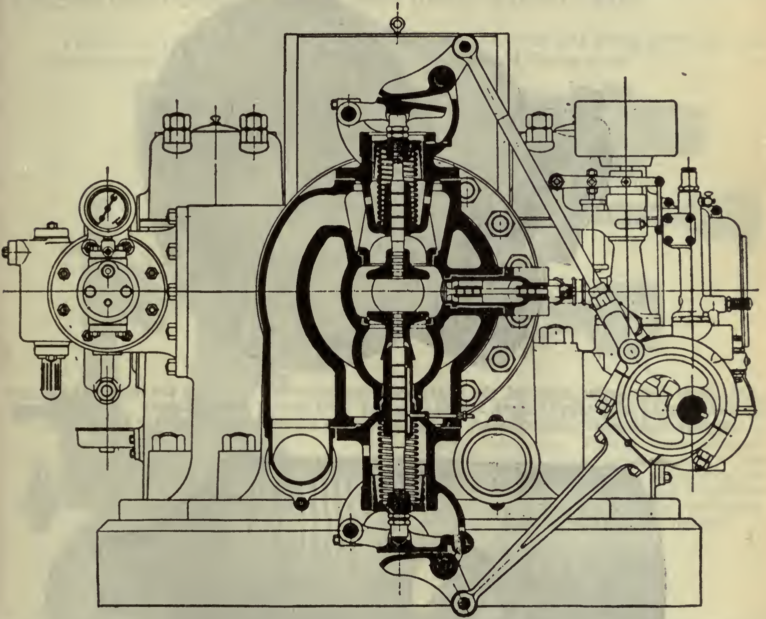


FIG. 2,880.—Transverse section through cylinder head of the National Diesel engine showing valves and valve gear. The inlet and exhaust valve seats are not cast in the head. The inlet valve seat is formed in the inlet cage and being located above the exhaust makes it possible to withdraw the exhaust valve by removing the inlet valve casing. Both inlet and exhaust valves are vertical. The valve heads are of cast iron, fastened to open hearth steel valve stems. Both valves are operated by rockers and wipers, which receive their motion from a single eccentric.

of the National engine show the construction and general arrangement of these parts.

Fuel Pump.—This is usually of the single acting plunger type, being driven by an eccentric or cam from the valve shaft. For multi-cylinder engines, a separate pump is provided for each cylinder. The capacity of the pump is made considerably more

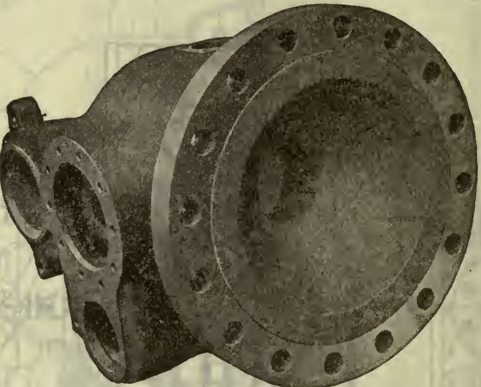


FIG. 2,881.—Cylinder head of the National Diesel engine. It carries the inlet, exhaust, air starter and fuel sprayer valves, and is provided with liberal water jackets. Note the substantial construction as indicated by the numerous holes in the flange for large bolts.

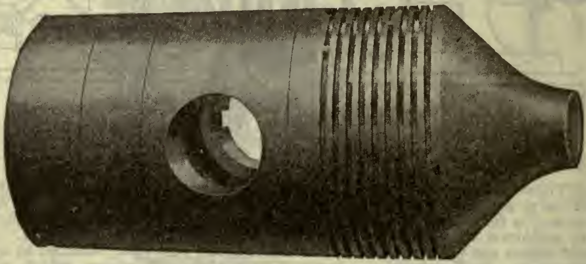
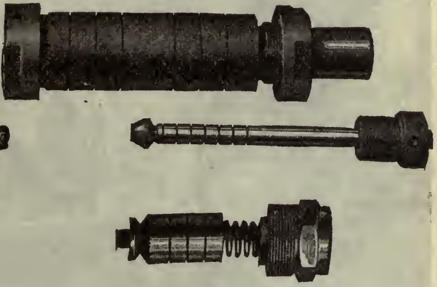


FIG. 2,282.—Piston of National Diesel engine. The piston is of the trunk type, of sufficient length to insure a low unit bearing pressure on the cylinder walls. Its length and the fact that nine rings are provided will indicate forcibly the severe conditions imposed by the Diesel cycle, requiring very massive and substantial construction.

than is necessary to allow for possible leakage combined with heavy overload.

Sprayer and Pulverizer.—The efficiency of the engine depends largely upon the effectiveness with which this device atomizes the fuel, as for proper combustion the fuel must be injected into the cylinder in a very finely divided state.

There are various types of sprayer and much time has been given to the development of this part as it is a vital element of the engine.



FIGS. 2,884 to 2,886.—Sprayer dissembled of National Diesel engine. The sprayer is of the open nozzle type, and is located in the extreme outer end of the combustion chamber. Fuel oil from the main fuel pump is deposited in the open channel of the sprayer body; as soon as the valve controlling the compressed injection air opens, this oil is injected into the combustion chamber thoroughly atomized, and is ignited by the heat of the compression.

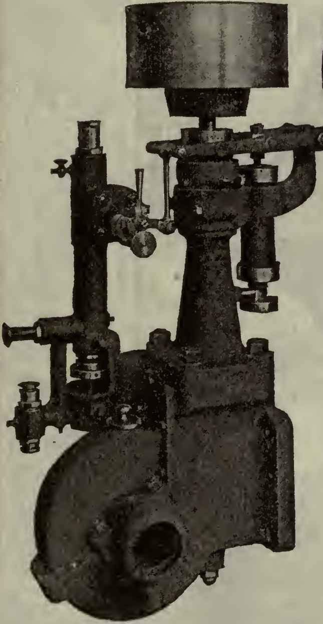


FIG. 2,883.—Governor and main fuel pump of National Diesel engine. These two devices are mounted together on the side of the main frame. The fuel pump is driven by an eccentric on the lay shaft. The pump plunger is of the differential type, the upper or larger part being hollow and having a cut off valve seated at its upper end. This cut off valve is under direct control of a Johns governor. The plunger has a full positive stroke with each revolution of the lay shaft. Governing is effected by allowing the cut off valve to seat at a predetermined point in the up stroke of the plunger, thus delivering to the sprayer a quantity of oil, correctly proportioned to the load which the engine is carrying. The governor is driven by spiral gears from the lay shaft.

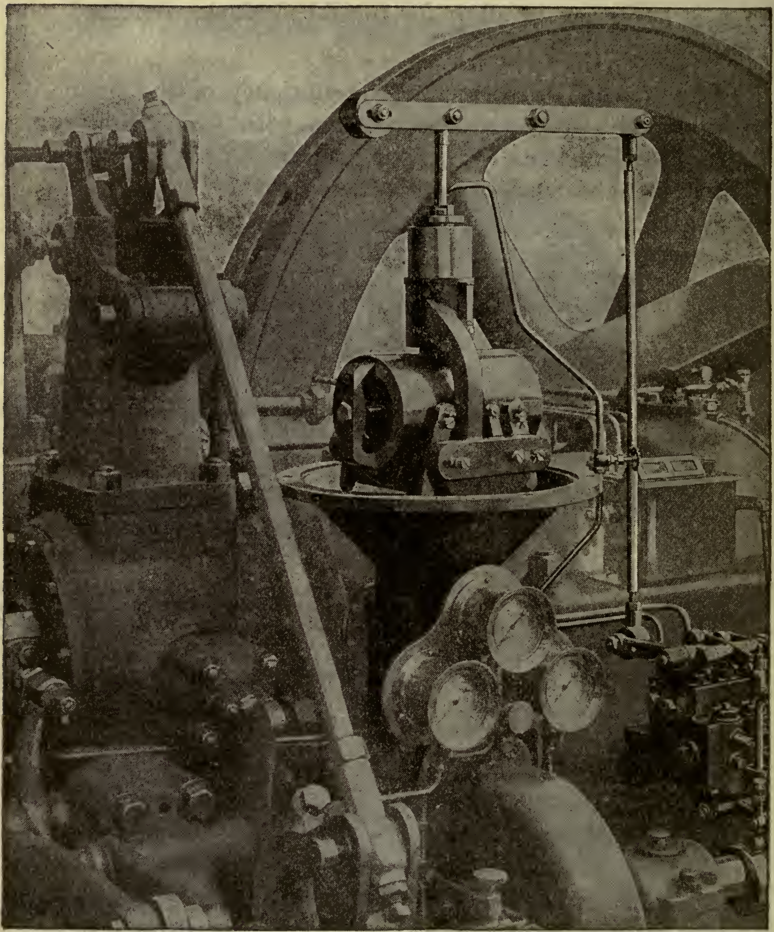


FIG. 2,887.—View of Allis-Chalmers Diesel engine showing governor with casing removed. The governor controls the amount of fuel oil supplied to the nozzle, this being accomplished by varying the effective stroke of the pump plunger by means of a by pass. A spring return on the suction stroke keeps the plunger always in contact with the roller on the driving lever, insuring uniform motion free from shock. Since the open nozzle does not require the oil to be delivered against pressure, the pump is simply a measuring device.

In one arrangement the movement of the fuel valve is very slight, giving a narrow annular opening for the entry of the oil. Surrounding the valve spindle is a series of brass washers perforated parallel to the spindle by numerous small holes. The oil is pumped into the space around the valve spindle near its middle, and by capillary action finds its way between the washers and into the perforations. The air for fuel injection is admitted through another pipe into the same space, but behind the oil, and because of its high pressure blows the oil in a very finely divided state into the cylinder when the valve opens.

Governor.—The amount of oil fuel injected into the cylinder is regulated by the governor which controls the time of opening of a by pass connecting the discharge and suction sides of the oil pump. At light loads the oil is pumped to the fuel valve for part only of the admission period, and air above will enter past the valve for the rest of the period.

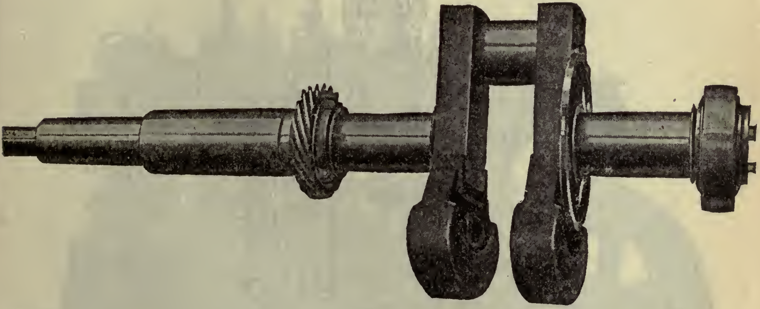


FIG. 2,888.—Crank shaft of National Diesel engine. It is made of open hearth steel, forged under hydraulic press. Counterweights of suitable proportions are bolted to the crank web and lubrication of the crank pin is effected by means of brass grooved centrifugal rings. The shaft is enlarged between the frame and outboard bearing for the flywheel fit. The lay shaft is driven from the crank shaft by means of cut spiral gears, one of which is seen in the illustration.

Starting Valve.—The engine is started by compressed air from an auxiliary reservoir, a special starting valve being used for the purpose.

Air Compressor.—For the operation of a Diesel engine a supply of air compressed to from 700 to 1,000 or more lbs. pressure is necessary. This high pressure puts severe duty on the compressor and accordingly it is an important part of the engine.

In medium size engines a two stage compressor is generally used, and for large engines the air is compressed in three stages, intercooling being provided so as to maintain a moderate temperature, and thus avoid explosions which might result from the lubricating oil.

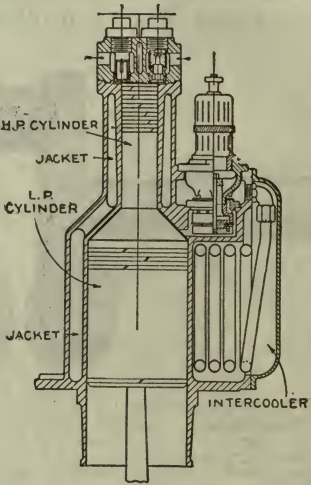


FIG. 2,889.—Fulton air compressor. It is of the two stage type and is driven directly from the main crank shaft. Free air is drawn into the larger cylinder and compressed to seven or eight atmospheres (approximately 105 to 120 lbs. per sq. in.) after which it is delivered to an inter-cooler before being drawn into the high pressure cylinder, where it is compressed to from 55 to 65 atmospheres (approximately 825 to 975 lbs. per sq. in.); thence to a second cooler, after which it is stored in the small air receiver for fuel injection. The compressor delivers slightly more air than is required for fuel injection and this surplus is stored in the two large receivers for starting. Air in the starting receivers may be at any pressure from 700 to 1,000 lbs. per sq. in., but the injection air should vary with the load, *i. e.*, 800 lbs. for half load, 975 lbs. for full load and slightly higher, say 1,000 lbs. for 10 per cent. overload.

The compressors are preferably directly connected to the crank shaft of the engine, though on some engines they are operated by a rocker arm from the connecting rod as well as by individual motion with the larger units. The power absorbed by the compressor is about 8 to 10 per cent of the total power of the engine.

Valve Gear.—The general arrangement consists of a long shaft driven from the crank shaft by means of spiral gears. It serves to drive the valve gear, governor, sprayer, and starter, lubricating and fuel oil pumps.

The general arrangement of the valve gear on both horizontal and vertical engines is shown in the accompanying cuts, also the various other details or mechanisms described in the preceding sections.

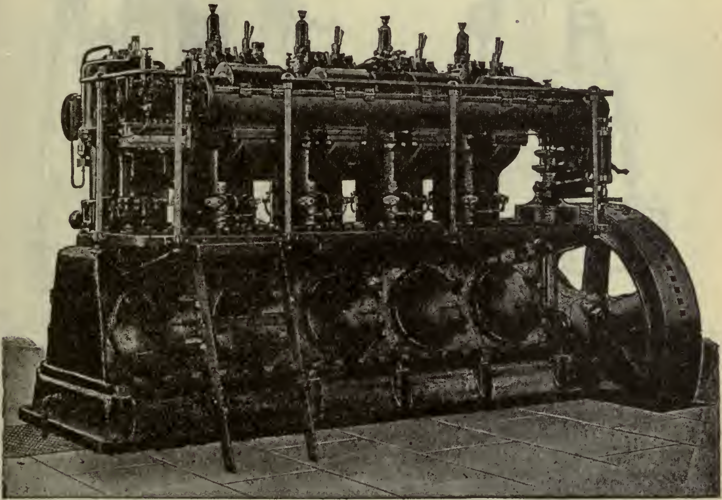


FIG. 2,890.—Busch-Sulzer Diesel engine, 120 brake horse power.

Lubrication.—On horizontal engines, the main heavy outboard bearings and cam shaft bearings are usually lubricated by means of chains running in oil pockets. The piston, piston pin, crank pin, eccentric, exhaust valve stem, and low stage compressor receive lubrication by force feed, minor bearings are lubricated by sight feed oil cups and compression grease cups.

A typical lubrication system for vertical engines would consist of a force feed oiler for the cylinders and piston crank shaft bearings. In one make of vertical engine the crank shaft is drilled through its entire length and a small self-priming rotary pump circulates the oil from the engine bed to all bearings located inside the crank case, and up through the hollow connecting rod to the wrist pin. The oil flows back to the crank chamber where it is filtered and so used over and over again. An overflow valve in the system regulates the amount of pressure which is normally held at 15 lbs; and a gauge mounted over the force feed lubricator may be readily observed by the operator.

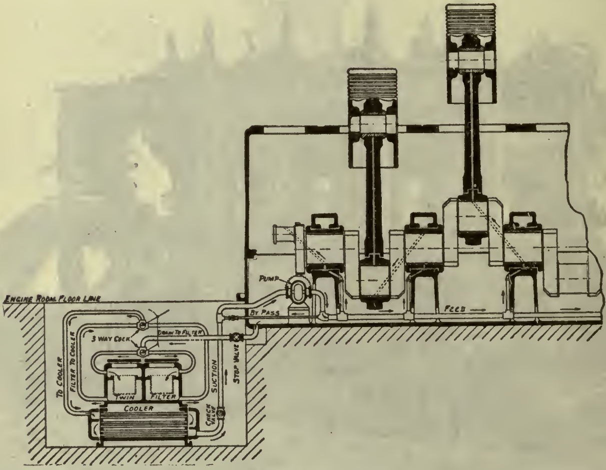
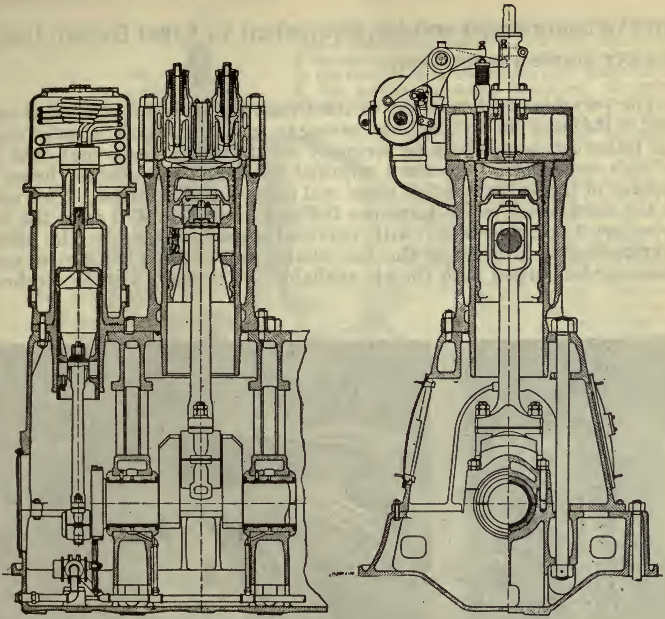


FIG. 2,891.—Schematic diagram of Busch-Sulzer-Diesel engine showing force feed lubrication system.

Economy.—The main feature of the Diesel engine is its economy, its thermal efficiency being variously estimated at from 32 to 34 per cent.

According to one American builder the fuel oil consumption for a Diesel engine of moderate size ordinarily runs about .43



FIGS. 2,892 and 2,893.—Sectional views of Busch-Sulzer four cycle Diesel engine. The cylinder heads contain all valves necessary to the cycle—that is, starting, fuel, admission and exhaust valves. The fuel valve is centrally located in the head. The starting and fuel valves are operated by levers fulcrumed upon an eccentric quill which can be rotated by hand lever so as to change the position of the valve levers relatively to their cams, the arrangement being such that, while the starting valve levers are in contact with their cams for starting, the fuel valve levers are out of contact with their cams. A turn of the handle, through a neutral position, reverses these conditions for running on fuel. One handle and one movement eliminate the factor of human error from this operation. The cam shaft is located alongside of and level with the cylinder heads. It is enclosed in a cast iron casing provided with hinged covers, the cams dipping in oil. It is operated from the crank shaft by a vertical shaft and two sets of heavy screw gears. All cranks are in one plane, the two center ones forming an angle of 180° against the two end ones. The multiple type fuel pump is operated from two eccentrics located on the vertical governor shaft. The pump gives two discharges of fuel for each power stroke, thus affording very accurate speed regulation, as a change of load between any two power impulses will influence the quantity of fuel pumped—the second discharge stroke just preceding ignition. The three stage air compressor has the general appearance of a fifth cylinder in the engine. It is driven from the crank shaft. Means of affecting the cooling of the compressed air between stages and after compression, and for the extraction of moisture are provided. The engine is equipped with a forced lubrication system operated by a rotary positive displacement pump, driven from the crank shaft. The system contains filters in duplicate and a cooler. The oil is fed at a pressure of 15 pounds. Suitably drilled holes, in the crank shaft and connecting rods, conveying the oil to the crank and wrist pin bearings, provide proper lubrication and cooling in these vital spots. After the system is once supplied, the oil is used over and over again. For lubricating the cylinders an adjustable, multi-feed, pressure lubricator is used. Both systems, being practically automatic, eliminate the personal factor of the operator.

pound in commercial service, equivalent to 8,000 British thermal units per horse power hour.

The very high fuel economy of the Diesel engine is about all that can be said in its favor for to offset the saving in fuel over other types of engine, the following items must be seriously considered: 1, very high first cost; 2, high compression pressure required to ignite the charge, hence any leakage of the valves, piston rings will cause faulty ignition. This applies to the compressor which furnishes the spraying air as well as to the power cylinder; 3, trouble likely with overload operation because the effect of overloading is to increase the fuel charge so that the amount is greater than can be burned with the air available. The result is that combustion continues after expansion has begun, and if the overload be great it may even be that the charge will still be burning when the exhaust valve is opened, which condition will quickly destroy the exhaust valve; 4, operation with leaky valve will result in both valve and seat being cut out by the hot gases blowing through at high velocities; 4, close and high grade attention

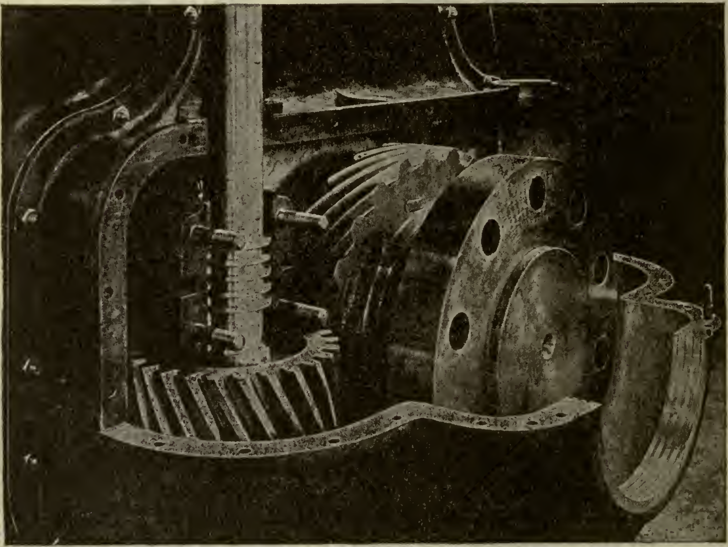


Fig. 2,894.—Vertical lay shaft and actuating screw gears of Busch-Sulzer Diesel engine.

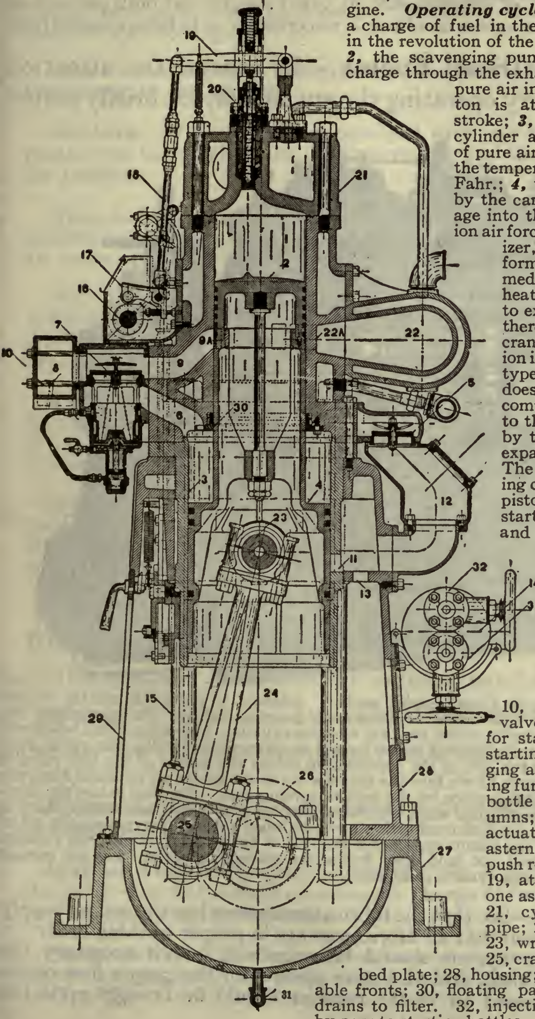


FIG. 2,895.—Southwark-Harris two cycle Diesel engine. ***Operating cycle: 1,*** the fuel pump injects a charge of fuel in the atomizer at a certain time in the revolution of the engine and leaves it there; ***2,*** the scavenging pump blows out the previous charge through the exhaust and leaves a charge of pure air in the cylinder when the piston is at the end of the outward stroke; ***3,*** the piston returns in the cylinder and compresses this charge of pure air to about 500 lbs. pressure, the temperature rising to about 1,000° Fahr.; ***4,*** the atomizer spindle is lifted by the cam shaft, opening the passage into the cylinder, and the injection air forces the oil, lying in the atomizer, into the hot charge in the form of a spray. The oil immediately ignites and further heats the air, and causes same to expand behind the piston and thereby transmit power to the crank shaft. There is no explosion in either this or the four cycle type of engine, the pressure does not materially exceed the compression pressure, but owing to the additional heat supplied by the burning of the fuel, the expansion creates the power. The parts are: 1, main or working cylinder; 2, main or working piston; 3, scavenging and air starting cylinder; 4, scavenging and air starting piston; 5, scavenging air inlet valve; 6, scavenging air outlet passage, starting air inlet passage; 7, scavenging air delivery valve; 8, scavenging air manifold between adjacent cylinders; 9, scavenging air inlet to working cylinder; 9A, scavenging air inlet ports; 10, air operated intercepting valve for air starting; 11, outlet for starting air; 12, silencer for starting air exhaust and scavenging air inlet; 13, vents for housing fumes; 14, injection air storage bottle; 15, removable front columns; 16, cam shaft; 17, atomizer actuating rockers (one ahead, one astern); 18, radius or atomizer push rods (one ahead, one astern); 19, atomizer levers (one ahead, one astern); 20, atomizer spindle; 21, cylinder head; 22, exhaust pipe; 22A, exhaust outlet ports; 23, wrist pin; 24, connecting rod; 25, crank pin; 26, crank shaft; 27, bed plate; 28, housing; 29, light sheet steel removable fronts; 30, floating packing rings; 31, crank pin drains to filter. 32, injection air from compressor; 33, by pass to starting bottles.

required; 5, high cost of maintenance; 6, great weight and bulk per unit of power developed; lubrication troubles; 7, uncertainty as to future cost of fuel

How to Run a Diesel Engine.—In general the attention necessary for properly operating the engine may be briefly stated as follows:

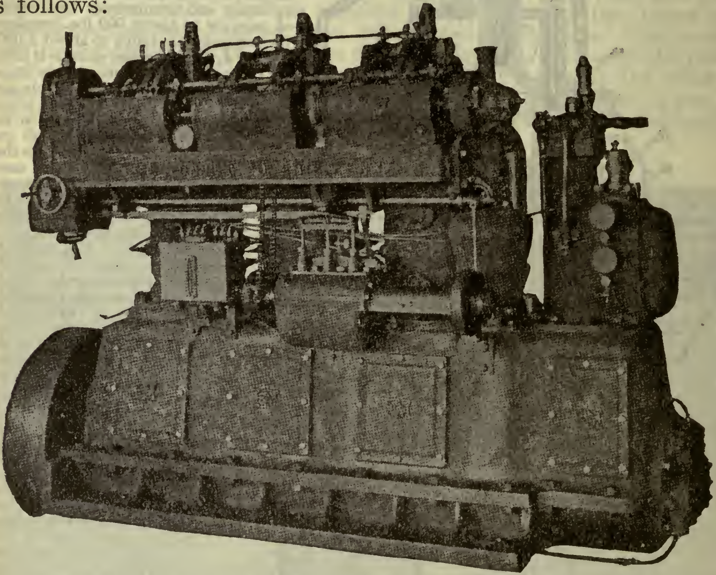


FIG. 2,896.—Fulton four cycle Diesel marine engine. Adapted to low grade residue or fuel oils but will operate also on kerosene. Compression pressure 500 lbs. ***In operation,*** the piston on the first down stroke draws pure air into the cylinder, with no mixture whatever on the second or return stroke this air is compressed to upwards of 500 lbs. per sq. in. Just before the piston reaches the end of the compression stroke the needle of the fuel valve is opened and the fuel charge is blown into the cylinder by the injection air obtained from the small receiver. Ignition takes place by the heat of compression. The fuel enters the compression space in the form of a fine spray and flame propagation takes place slowly; there is no sudden explosion, but gradual combustion and continued pressure for practically all of the working stroke. As the piston approaches the end of the working stroke, the exhaust valve in the head is opened and held open during the scavenging stroke, and the burned charge is forced out, thus the cycle is completed.

Before Starting.—See that the lubrication system has the proper supply of oil, care being taken that all lubricators are in proper working order.

The cooling water system should be examined, and if necessary the circulating pump primed, and all valves opened for the proper flow of the water. The compressed air in the receiver should be brought up to the proper pressure.

Place engine crank in proper position for starting, exhaust cam on low compression side, and place control handle of starting valve in proper position for starting.

Open air inlet to low pressure side of the air compressor. Pump by hand fuel charge to the sprayer.

Starting.—If the engine be provided with automatic starting arrangement, open the valve on the starting receiver and the valve on the injection air receiver.

When engine comes to speed, put starting valve lever in the off position and shut off receiver containing the starting air, also open full the injector air valve.

Engines may be started with air at pressure as low as 200 to 300 lbs. If, however, the injection air pressure be below 600 lbs., the valve on the injection air receiver should be closed when the starting air receiver is opened. It can be reopened as soon as the engine has come to speed.

Running.—If light and heavy fuels are used, the heavy fuel can be turned on after starting, being careful if the heavy oil requires heating not to admit the heavy fuel until the engine has heated up sufficiently to raise the temperature of the jacket water to about 110° Fahr.

Stopping.—To stop the engine shut off the fuel supply and close tightly the injection air receiver valve. In freezing weather the jackets should be drained and the cooling air coils between the high pressure air and intermediate compressor should be blown out as well as the injection pipe.

Diesel Engine Troubles.—Like all other types of gas engine, the Diesel engine is subject to various troubles due principally to careless or improper attention from those in charge. The following are the most frequently met with:

Failure to Start.—Usually due to faulty working of starting valves and failure to ignite.

Ignition Fails.—The principal cause is loss of compression. Sometimes the pump does not deliver an adequate charge of fuel, or the timing of the fuel injection may be incorrect.

Pump Fails.—The check valves may be prevented seating by foreign matter in the oil, or the pump may be air bound. The connections and stuffing box must be tight to prevent leakage.

Loss of Compression.—This may occur by leakage, past the pisto due to stuck or worn piston rings, or past the valve which may need r grinding. Leakage past the exhaust valve may be detected by the col of the exhaust gases; if smoky or black, it indicates that the gases a leaking past the valve, resulting in improper combustion. Owing to tl very high temperature of the Diesel cycle any leakage past the exhau valve if allowed to continue will very quickly ruin not only the valve b the valve seat.

Overloading.—Since the governor regulation is such that it increas the fuel charge with an increase in load, when the engine is overloade the temperature of the exhaust gases is very high, and in some cases exhau takes place with combustion still in progress. No exhaust valve will star such temperature for any length of time, and for this reason a Dies engine should not be overloaded.

Underloading.—Because of the governor action in varying the quanti of fuel charge with the load, ignition may cease in one of the cylinders a multi-cylinder engine because of lack of fuel supply.

Irregular Running.—This may be due to several causes, as defecti governor action; fuel injection too early or too late; individual pumps multi-cylinder engines not delivering like charges—that is, one pump ma deliver more fuel per stroke than the other, due usually to improper acti of the check valve or air pockets.

CHAPTER 49

GAS PRODUCERS

Producer Gas.—The gas formed by a producer is the product of an incomplete or retarded combustion of the fuel. This is effected by burning the latter in a bed, *several feet in thickness*, within the *generator* or producer proper. The heated gases and flame from the portion nearest the air supply, where combustion is more or less rapid and complete, have to pass through the remainder of the bed, distilling gases from the latter in their transit. The processes of combustion within the generator may be described as follows:

The oxygen O, of the atmosphere combines with the fuel in the first portion of the bed, and the ensuing combustion results in the formation of carbon dioxide CO_2, that is to say, each atom of carbon combines with two atoms of oxygen. This means that each pound of carbon requires 2⅔ pounds of oxygen, the combination resulting in the formation of 3⅔ pounds carbon dioxide, and, at the same time, imparting about 14,500 heat units to the mass of fuel.

The effect of this process is that the 3⅔ pounds of carbon dioxide assimilate another pound of carbon, forming 4⅔ pounds of carbon monoxide, which is equivalent to the evolution of 2⅓ pounds of the latter gas for each pound of carbon consumed.

Processes Within the Producer.—The first two reactions in the generator constitute what is termed incomplete combustion, consisting of the formation of ***carbon dioxide*** and the decomposition of the compound into the ***monoxide.*** The result of the process is the same as if the pound of carbon had been burnt directly to carbon monoxide, thus developing 4,451 *B.t.u.* for each pound of C, burnt to CO. This heat is apparent as the sensible heat of the fuel bed and generator.

The carbon monoxide may be caused to combine with fresh oxygen at any time after it has left the producer, this second combustion to CO_2 being accompanied by the evolution of $14{,}544—4{,}451=10{,}093$ *B.t.u.*, the heat value of CO to CO_2 per pound of carbon contained.

Now, atmospheric air is a mechanical mixture of oxygen, which supports combustion, and of nitrogen, which is an inert gas serving as a diluent. The proportions, by weight, are: oxygen, 23 per cent.; nitrogen, 77 per cent. Therefore, accompanying and diluting the 2½ pounds of carbon monoxide, resulting from the reaction in the producer, there will be about 4.46 pounds of nitrogen, the whole forming about 6.8 pounds of lean gas.

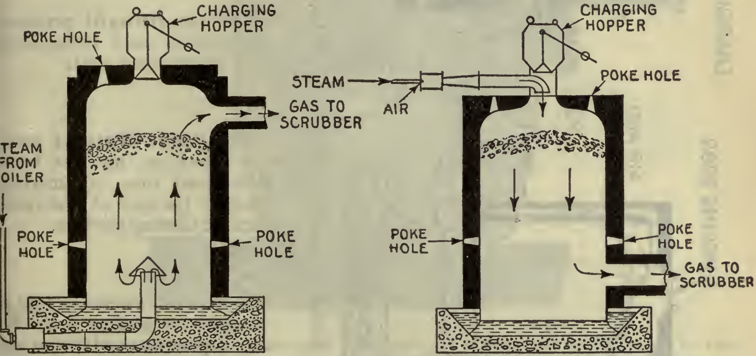
Use of Steam in Connection with Gas Producers.—The object of the gas producer being the generation of a combustible gas of high heat value, it is desirable to use as little as possible of the inherent heat energy of the fuel during its gasification. The only sensible heat necessary is that which raises the fuel to incandescence so as to reduce the carbon dioxide to monoxide.

Now, there are some 4,451 heat units generated in burning a pound of carbon to carbon monoxide, and as only part of this heat passes away with the gases or is dissipated by radiation from the generator, it may be said that there is a surplus of about 2,500 *B.t.u.* per pound of carbon over and above what is necessary for the working of the furnace. This extra heat requires to be utilized in order to get the greatest efficiency from the generator and to keep the temperature down, a temperature of 1,900° Fahr. having been found to give a minimum of CO_2 or a maximum of CO.

This surplus of heat is utilized in an effective manner by the introduction of steam into the generator.

Each pound of steam on being decomposed, liberates ⅑ pound hydrogen and ⁸⁄₉ pound oxygen, the latter combining with ⅔ pound carbon to form 1⁵⁄₉ pounds of carbon monoxide, this extra amount being produced without the introduction of any more inert nitrogen into the apparatus.

In order to effect the decomposition of one pound of steam, in the pressure of incandescent carbon, 6,800 heat units are necessary, and as there are only about 2,500 *B.t.u.* available for the purpose from the burning of each pound of carbon, the quantity of steam that can be taken care of is $\frac{2,500}{6,800}$ or about .37 pounds. This yields about .325 pound of oxygen which combines with about .245 pound of carbon, yielding .57 pound of the monoxide, which is equivalent to saying that about one-fifth of the total carbon can be gasified by the aid of the steam. The hydrogen liberated will be about .033 pound for each pound of total carbon gasified in all reactions.



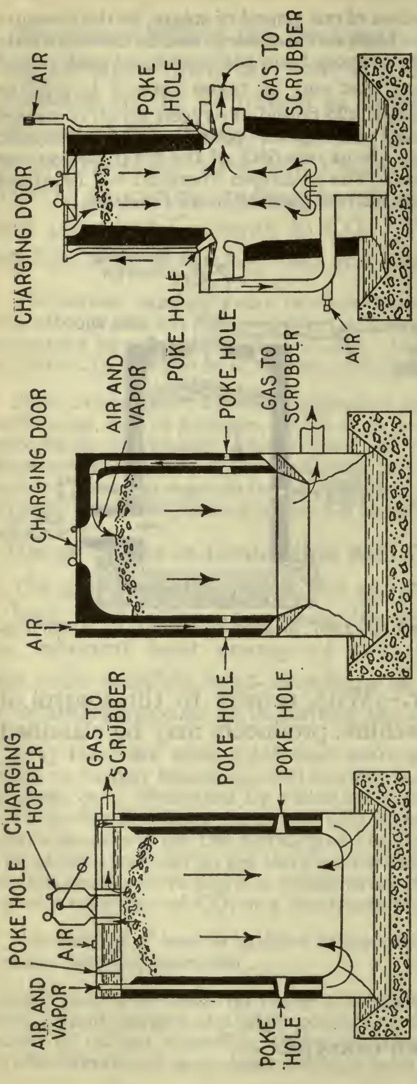
IGS. 2,897 and 2,898.—Gas producer types; fig. 2,897, pressure up draught; fig. 2,898, pressure down draught.

Classes of Gas Producer.—With respect to the control of he air passing through the machine, producers may be classified s:

1. Pressure producers;
    a. Up draught;
    b. Down draught.

2. Suction producers;
    a. Up draught;
    b. Down draught.

3. Combined up and down draught.



FIGS. 2,899 to 2,901.—Gas producer types; fig. 2,899, suction up draught; fig. 2,900, suction down draught; fig. 2,901, suction two zone producer.

In the ***pressure system,*** the air required for the generation of the gas is delivered to the gas producer under pressure. derived from an auxiliary source, and the gas generated in the producer is delivered to the engine under the same pressure.

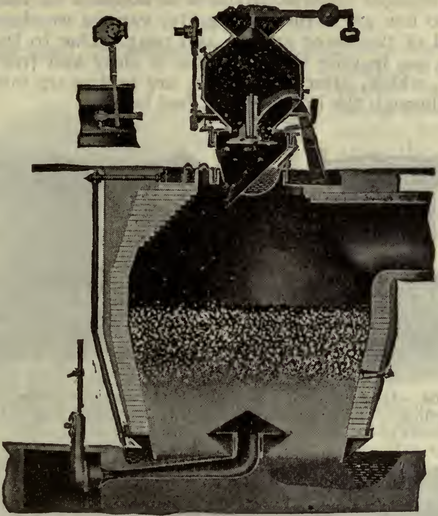
The ***up draught*** pressure producer is used principally with bituminous coal.

The ***down draught*** pressure producer has not been introduced into commercial practice because of the complicated feeding devices.

Moreover, whenever fuel is changed or stoking is necessary it interferes with the gas production and causes irregularity in the operation.

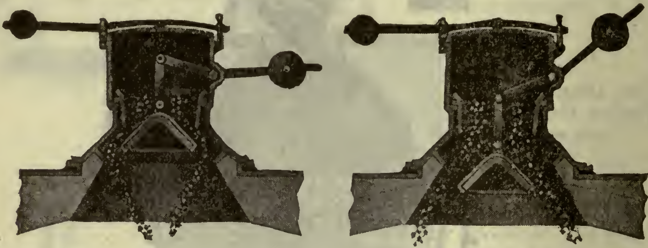
In the ***suction system,*** both the passage of the air through the producer and the introduction of the gas generated therein into the cylinde of the engine is effected by the sucking action of the pisto during its forward or chargin stroke.

For anthracite coal an high grade of charcoal an coke the up draught suctio producer is used and may b considered a standard type.



. 2,902.—Morgan pressure up draught producer with George automatic feed. In construction the steel shell is contracted toward the bottom edge, which extends into a water-illed ash pan formed as a depression in the foundation. The shell is supported by four ast iron feet resting on the concrete floor surrounding the ash pan. About three feet above he water level in the ash pan a number of sight holes are provided around the circumference f the producer for observing the fuel level and combustion. The producer top is covered y a shallow thick steel pan filled with water. Through a central opening the automatic eeding apparatus communicates with the interior of the producer. Poke holes closed by vater sealed covers are placed around the circumference of the pan. The ***blowing device*** onsists of one central *tuyere* with mushroom shaped top, which, rising from the ash pan, listributes a mixture of air and steam uniformly over the interior of the producer. The last is controlled by an *injector*, constructed to give the proper proportion between steam and ir at all pressures. In small plants it is preferable to have the steam supply to the blower nder the direct control of the operator at the furnace, who is able to judge the quantity of as which at any time may be required. The make of gas is entirely controlled by adjusting 1″ steam valve. The ***continuous automatic feeding device*** consists of an inclined lowly rotating water cooled feeding spout, designed to continuously feed over the entire urface of the fire. ***In operation,*** the layer of ashes should extend to a height of about ft. above the ash pan, or to such a point that the top layer of ashes is kept opposite the ight holes in the shell. Thus the tuyere is surrounded by ash and protected from the heat. horizontal layer of incandescent coal from 2 to 3 ft. thick commences at the level of the sight oles. In the upper layers of the coal a somewhat lower temperature prevails. The amount f steam required is from 33 to 40% of the weight of the fuel gasified.

The down draught suction producer is adapted to various bituminous coals, and also low grade fuels as lignite, various woods, refuse, bagasse, coke, charcoal of the lower grades, and peat. Due to the down draught principle, the gas leaving the generator is clear and free from all heavy hydrocarbons, which, after distillation, are broken up into fixed gases by being drawn through the incandescent fuel bed.



Figs. 2,903 and 2,904.—Hilger charging hopper. ***In operation*** coal may be charged towar the center by opening hopper valve slightly as in fig. 2,903, or toward the circumference b opening valve wide as in fig. 2,904.

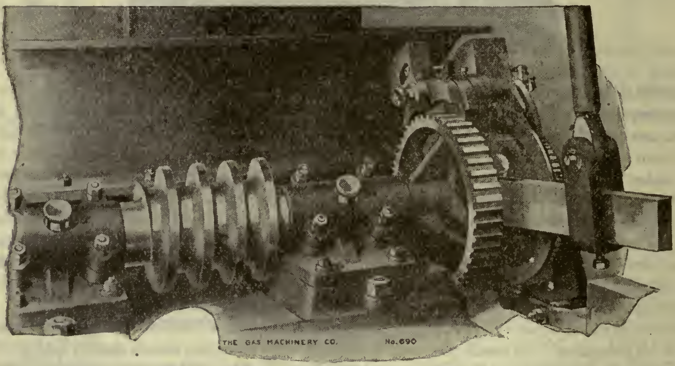


Fig. 2,905.—Hilger "Pilgrim" step motion for rotating the grate. The ash pan is rotated b worm gearing, first a desired distance in one direction, then back again for a somewhat small distance. This reversal is automatically accomplished by the flapping over of a dog, th mechanism being plainly shown in the cut.

Although the pressure producer is usually composed of more cumbrou apparatus, and requires more space for its installation, it possesses great elasticity than the suction system for meeting variations in fuel condition and has greater capacity for utilizing different kinds and cheaper grades fuel. It is undoubtedly the better for use in connection with large pow units, and in cases where several gas engines are operated from the same producer plant.

On the other hand, for isolated plants of small capacity, or where only a single gas engine of medium power is used intermittently the application of the suction system not only enables the simplification of matters relative to bulk and the reduction of cost, but it affords the more important advantage of making the demand of the engine for gas the controlling factor in the generation of the gas from the solid fuel.

The combined up-and-down draught producer is virtually a down draught suction producer placed on top of an up draught suction producer the gases being taken off at the center. An objection to this type is that more skilled labor is required to secure satisfactory results than the other type, the variations in the load and fuel requiring very close attention.

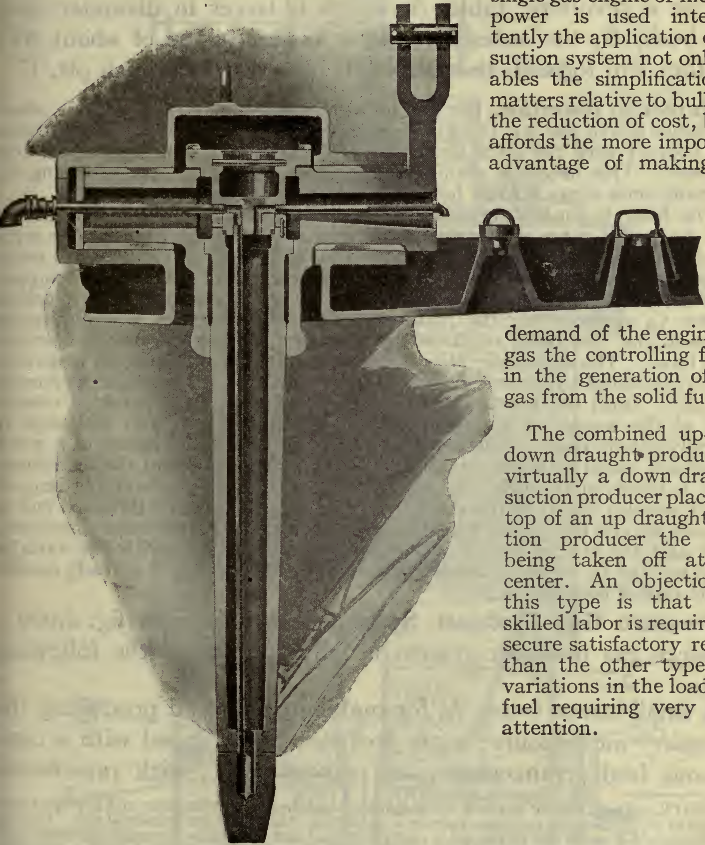


FIG. 2,906.—Hughes water cooled poker. The poker is suspended from the cover of the producer and extends into the fuel mass penetrating through the green fuel zone into the incandescent zone. ***In operation,*** the poker stirs the fuel constantly, swinging back and forth in a vertical plane as the fuel mass revolves, the point of the poker describes a series of loops. The path of the poker is from the center to a point a few inches from the brick lining of the shell.

How a Producer Operates.—Fig. 2,907 shows a section of a typical producer of the pressure type using anthracite coal The incandescent fuel is supported upon a bed of ash, this resting on a revolving table, A, which is larger in diameter than the bosh, B, the ashes forming a natural slope of about 55° As they are formed, the ashes fall into the sealed ash pit, C.

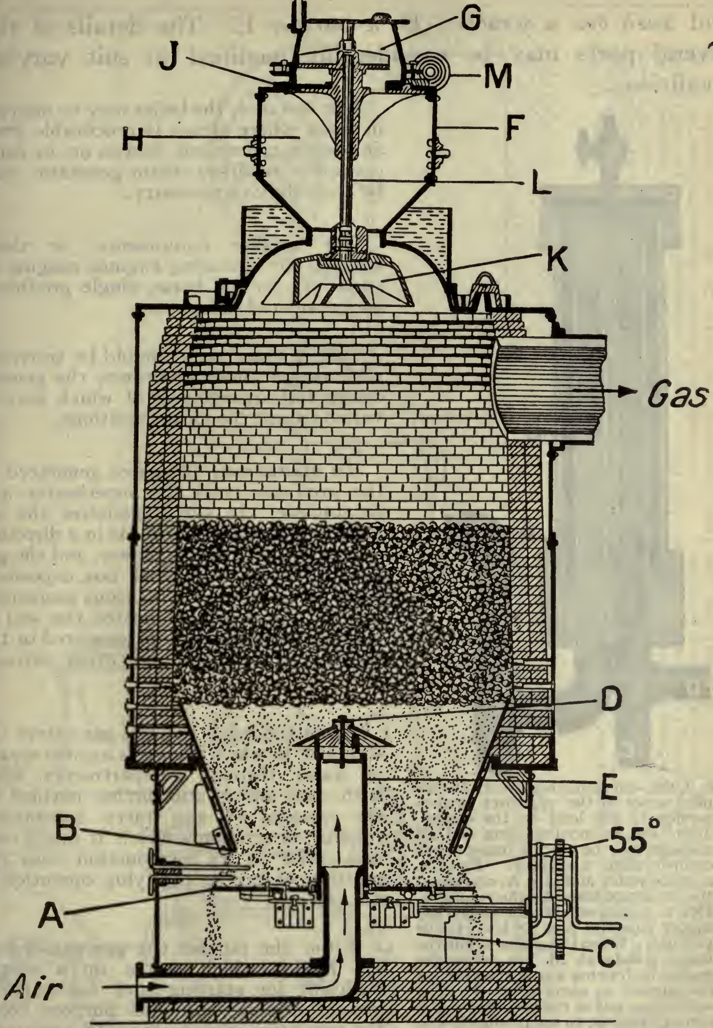
In regular operation, the line between the ashes and fuel is kept about six inches above the cap D, of the central air pipe E, so that the fire comes into contact with the brick lining only, all iron work being kept away from the heat. This height is maintained constant by grinding or revolving the ash table once every 6 to 24 hours according to the rate of working.

The blast is usually furnished by a steam jet blower, but a fan blower may be used if more convenient, a small steam pipe being run into the vertical air pipe to supply the steam required for softening the clinkers and maintaining the proper temperature of the producer. This producer is equipped with a continuous *automatic feed device* F, which consists of a *receiving hopper* G, surmounting the main *storage magazine* H, the communication between the two being regulated by a horizontal *rotary register* J, operated by a lever. The *distributor plate* K, is suspended below the main magazine and is supported by a steel shaft L, which passes upward through the storage cylinder. Both the hood of the distributor plate and the inverted conical base of the magazine are water cooled, thus facilitating the discharge of strongly caking coals. The receiving hopper is rotated by means of a worm wheel and worm attached to the upper end of the shaft, and the distributor plate is revolved through the radial arms and hub of the receiving hopper which are also keyed to the shaft. A *hand wheel nut* on the threaded end of the axis affords the means for adjusting the distance between the distributor plate and the coal reservoir. This adjustment, together with the variable speed secured by means of the step cone pulley M, permits of a ready control of the rate of coal discharge.

Pressure Gas Producer System.—As shown in fig. 2,909 a pressure gas producer system usually consists of the following parts:

A small steam *boiler*, A, for making steam and producing the necessary air pressure; a gas *producer*, B, equipped with a continuous feed arrangement; an *economizer*, C, with *superheater*

NOTE.—Demands for gas fuel are becoming more and more numerous and producer gas, because of its evident advantages, is surely supplanting other gases for industrial use. Coal or coke suitable for use in gas producers is easily obtained and the plants for gasifying these fuels are simpler in operation and less costly for installation and maintenance than the plants for gasifying most of the other fuels which have been used for gas making. Any gas supply to be satisfactory to the present day manufacturer must be efficient and dependable. Variations in the quality of the gas and fluctuations in the supply of gas both cause trouble. Hence it is highly desirable that a gas making machine shall produce gas of uniform quality steadily day after day. Such performance can result only from the following of correct mechanical principles and the use of excellent materials in the construction of the producer.



IG. 2,907.—Cross section of a typical gas producer of the pressure type using anthracite coal, as described in the accompanying text.

and *wash box* a *scrubber*, D, a *purifier* E. The details of the several parts may be considerably modified to suit varying conditions.

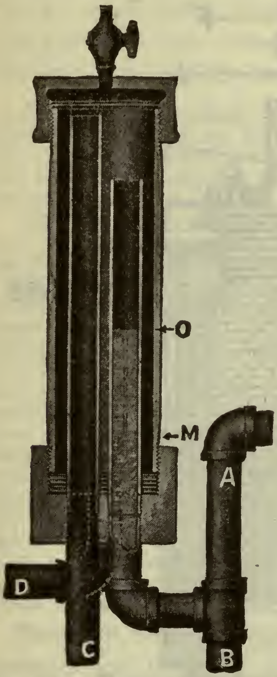


FIG. 2,908.—Syracuse regulator. It supplies water to the vaporizer in proportion to the load on the engine. There are no moving parts and all passages may be quickly inspected or cleaned when necessary. Pipe B, supplies water and pipe A, carries off the excess not taken by the regulator. Pipe C, is connected to the engine gas supply pipe and pipe D, leads to the producer. When the engine is idle the water stands at M, but when the engine is drawing gas it rises due to the suction to some point O. The water runs out of the slot shown and through pipe D, to the producer. The amount of this water is proportional to the load on the engine.

For instance, the boiler may be omitted in cases where steam is procurable from any other convenient source, or, in some cases, the separate steam generator may be absolutely unnecessary.

For smaller equipments, or those suitable for operating engines ranging in power up to 500 horse, single producers are considered sufficient.

Larger equipments should be provided with two or more producers, the general design and arrangement of which may be varied to suit the local conditions.

In operation, the gases generated in the producer enter the superheater and economizer. In the economizer the air blast of the producer travels in a direction opposite to that of the blower, and the gas passing through the wash box deposits a large portion of its extraneous suspended matter. Here also is located the seal or non-return against the gases stored in the holder and present in the other parts of the system.

From the wash box the gas enters the scrubber, in which it travels against sprays of water through compartments filled with coke and is still further purified by the removal of any tarry substances, sulphur, or ammonia which it might contain, prior to its introduction into the purifier where the purifying operation is completed.

From the purifier the gas passes into the holder which stores up a supply sufficient for starting and for running several minutes, its chief purpose being the regulation of pressure and variations in the consumption and mixture of gases.

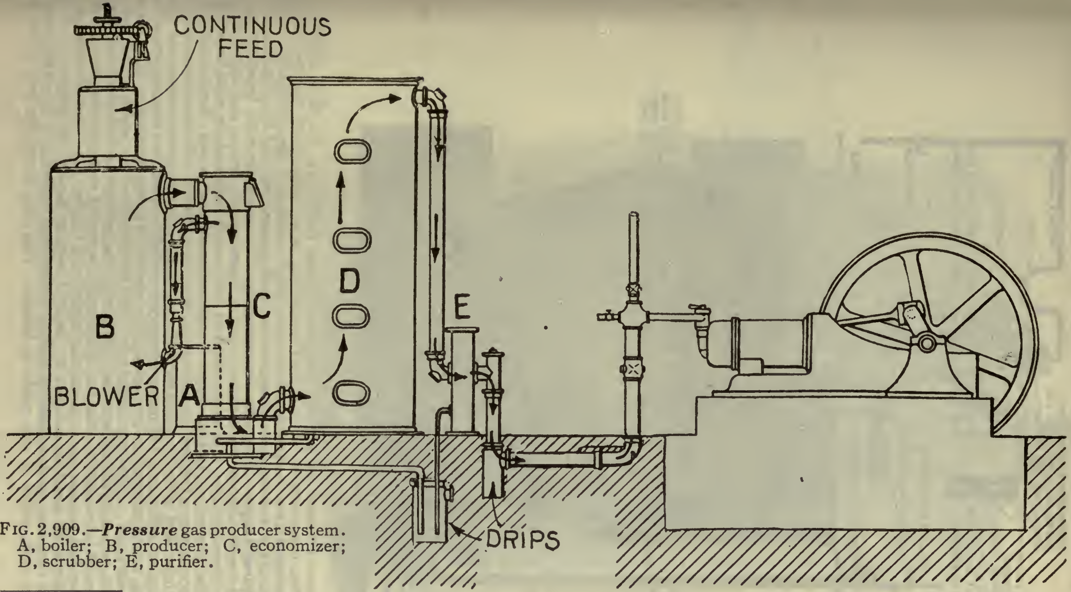


FIG. 2,909.—***Pressure*** gas producer system. A, boiler; B, producer; C, economizer; D, scrubber; E, purifier.

NOTE.—***The pressure producer*** was invented and developed long before the suction producer came into use. In this system air is forced into the producer by either a jet of steam or a blower, under a pressure of a few ounces per square inch. After the gas leaves the producer it is cooled, scrubbed and purified, and then passed into a gas holder of considerable size, from which source it is drawn to the engine for consumption. The gas is permanent and fixed and may be economically transmitted any distance for power and metallurgical purposes. In one class of pressure producer the valuable by-products of soft coal, tar and ammonia are separated during the process of gasification and form a valuable commercial asset. The latter system of production is called "the recovery process of pressure producers." The ordinarily well designed pressure producer will operate on a thermal efficiency of 80 per cent, but if the exhaust gases from the engine be utilized to the full extent in preheating the air entering the producer, a thermal efficiency of 90 per cent may be obtained.

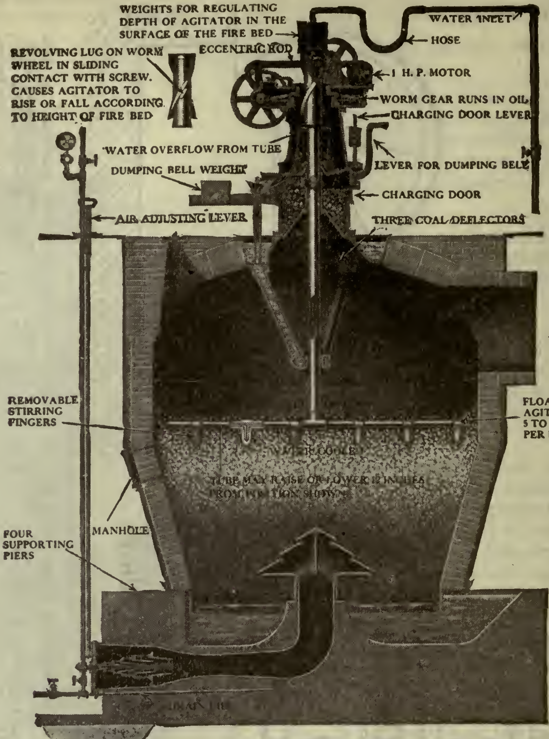
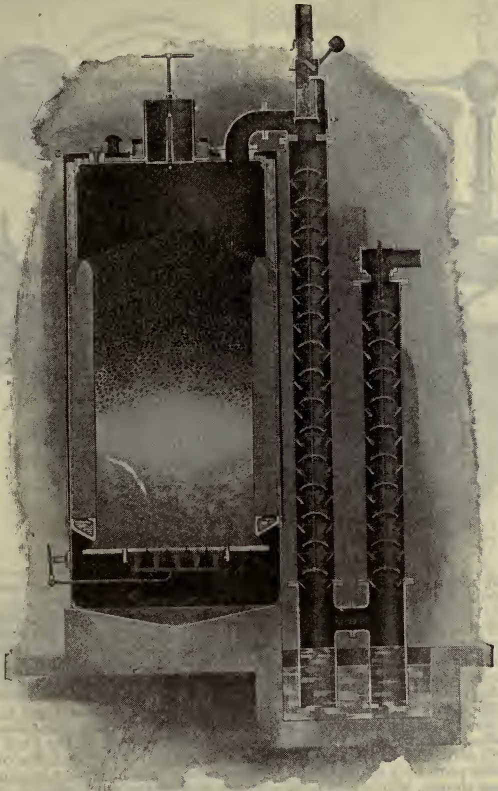

WEIGHTS FOR REGULATING DEPTH OF AGITATOR IN THE SURFACE OF THE FIRE BED
WATER INLET
HOSE
REVOLVING LUG ON WORM WHEEL IN SLIDING CONTACT WITH SCREW. CAUSES AGITATOR TO RISE OR FALL ACCORDING TO HEIGHT OF FIRE BED
ECCENTRIC ROD
1 H. P. MOTOR
WORM GEAR RUNS IN OIL
CHARGING DOOR LEVER
WATER OVERFLOW FROM TUBE
LEVER FOR DUMPING BELL
DUMPING BELL WEIGHT
CHARGING DOOR
AIR ADJUSTING LEVER
THREE COAL DEFLECTORS
REMOVABLE STIRRING FINGERS
FLOATING AGITATOR 5 TO 10 REV. PER HOUR
WATER COOLED
TUBE MAY RAISE OR LOWER 12 INCHES FROM POSITION SHOWN
MANHOLE
FOUR SUPPORTING PIERS

FIG. 2,910.—Chapman pressure up draught semi-mechanical producer. Its chief feature is the floating agitator, consisting of water cooled horizontal arms with short vertical strong fingers. The floating action is similar to the compound motion of the "Yankee" screw driver. The vertical shaft of the agitator is provided with a special driving head, having two large screw-like spiral flanges. These flanges engage in sliding contact with two lugs projecting inwardly from the hub of the driving wheel. Normally the wheel and agitator revolve together at the rate of five or more revolutions per hour, but should the agitator strike an obstruction or become submerged too deeply in the surface of the fire bed, it automatically "screws up" at once to a point where the forces are again in balance. An inner tube carries cooling water to the end of each stirring-finger, thus providing positive circulation, and also a means for forcing out sediment. If necessary, steam could be used for blowing out. The quantity of cooling water required is from one-half to one cubic foot per minute. The worm and worm wheel run in oil. The jack shaft is provided with chain oiling and large oil wells. The lower end of the stem of the agitator is held in place by a water cooled cast steel ring. This bearing is amply lubricated by the tar which collects on the water cooled parts. All of the cooling water has to pass down one of the legs of this bearing, and up the other two. This makes a sufficiently rapid circulation to prevent serious deposit of sediment. The dumping bell is of cast steel and is made in halves to facilitate replacing. It rests against a cast steel seating ring, which is also made in halves. The ring about the bell is water-cooled.



IG. 2,911.—Syracuse gas producer. ***It consists of*** a shell lined with fire brick and containing a hot, deep fire. An opening near the bottom lets in a mixture of air and steam and an opening near the top takes off the gas. The gas may be *drawn off*, causing a flow of air and steam into the bottom as in the suction type, or the air and steam may be *blown in*, causing the gas to flow off as in the pressure type. All the other parts of a producer such as charging hopper, grate, ash pit, steam generator, purge valve, scrubbers, etc., are auxiliary to the above. ***In operation,*** a fire is built as in an ordinary furnace, the draught being furnished by a blower or exhauster operated by hand or power. After the fire is well started, steam is mixed with the entering air and soon a combustible gas is being generated. When the gas burns well at a test cock near the engine, the blower or exhauster is stopped and the engine started. The generation of gas now proceeds automatically by the drawing in of the air and steam through the fuel bed and the resulting gas through the balance of the system, as a result of the suction caused by the engine. ***The scrubbers,*** cool and clean the gas. In the suction plant the gas is taken from the latter into the engine cylinder and no storage is necessary. Upon each suction stroke of the engine, gas is drawn into the cylinder and off the top of the producer, air and steam entering at the bottom to take its place.

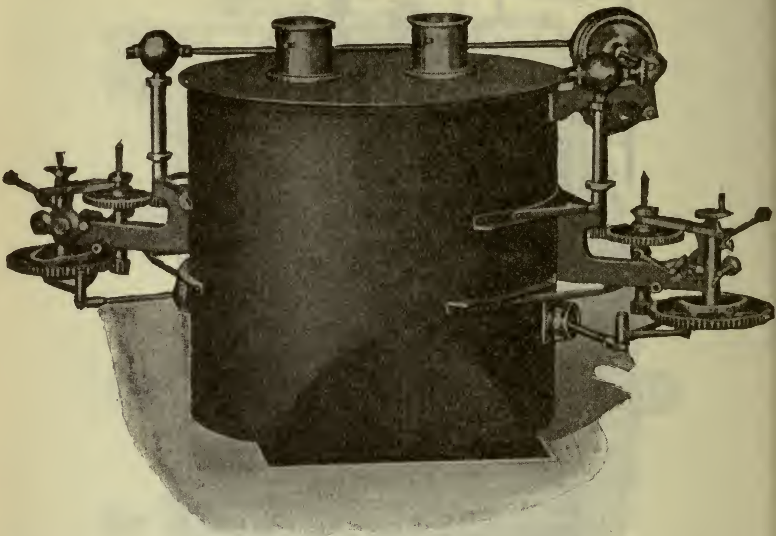


FIG. 2,912.—Treat mechanical gas producer. ***In construction*** the machinery is attached directly to the shell of the producer making a self contained unit. Two pokers are used for stirring the fuel, these acting at intervals of half to three-quarters of an hour alternately entering through the side walls, at diagonally opposite corners, slice through the fire in a horizontal plane, the combined effect of the two being to sweep through and cover practically the entire working area enclosed within the lining walls in a single movement. The movement is completed coincident with the complete withdrawal of the poker to a position outside the producer, where it remains at rest till it is time for it to repeat the movement, which occupying but half a minute, no time is afforded for it to get red hot, and hence no special arrangements are required for keeping it cool. As will be seen, the poker slides in and out of the producer through a large water cooled ball which rotates in a socket fixed to the shell of the producer, and serves as a fulcrum upon which it swings as it slides, the outer end being carried in a circular path by a crank having a large radius, the inner end is caused to describe the peculiar figure shown, which practically covers half of the working area of the producer as mentioned above. Since the crank may be revolved in either direction, the mechanism may be reversed at intervals to equalize the bending of the poker, otherwise, to carry the coal to one side of the producer. The crank action is effected by a heavy corrugated internal gear wheel having a crank pin journaled in a bracket fixed to its rim, to the lower end of which the end of the poker is loosely coupled, one complete revolution of this wheel produces one movement of the poker, the same being driven by a gear and clutch mechanism analagous to that employed in punching machinery, the wheel is set in motion, by an automatic trip gear, which has a time control, is automatically stopped by a cam fixed to its rim, which at the same time sets the tripper. The pokers are formed of plain round rough steel bars, and except for a hole drilled in one end, and a spiral groove cut in their surface, are not machined. ***In operating,*** following a poker movement, when the driving crank comes to rest, the inner end of the poker which remains resting in the water cooled ball, keeping the opening closed, is ready to reenter the producer, and at the same time is kept cool by the influence of the ball. As the poker is standing for eighty per cent of the time outside of the producer, and in a position easily accessible, it is easily and quickly uncoupled from the crank pin by simply pulling a pin; a strong laborer, unaided, being able to effect a change of pokers in a couple of minutes.

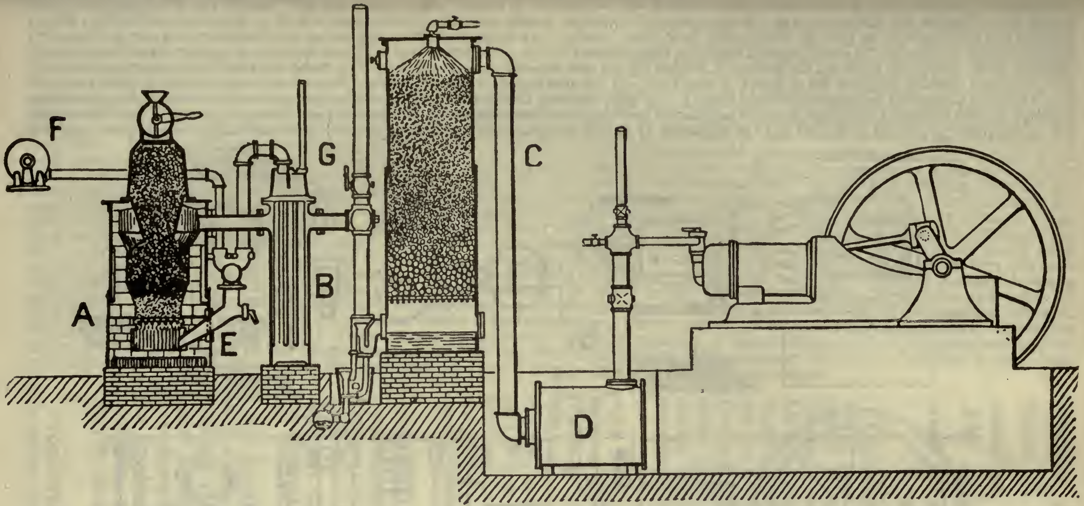
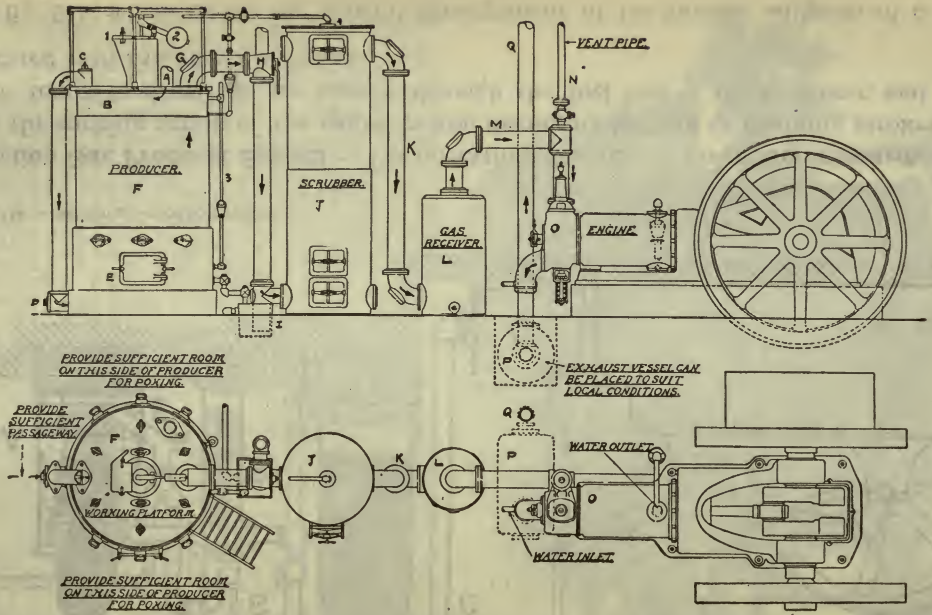


FIG. 2,913.—Suction gas producer system.

Suction Gas Producer System.—The operation of a suction gas producer system depends upon the sucking action of the engine piston during its forward or charging strokes, which action tends to draw the air supply through the fuel bed of the producer, and the gas generated into the engine cylinder.

In fig. 545, which shows the general arrangement of the various adjuncts of a suction


VENT PIPE.
N
Q
M
K
1
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C
G
H
A
B
PRODUCER.
F
3
E
D
I
SCRUBBER.
J
GAS RECEIVER.
L
ENGINE.
O
P
EXHAUST VESSEL CAN BE PLACED TO SUIT LOCAL CONDITIONS.
PROVIDE SUFFICIENT ROOM ON THIS SIDE OF PRODUCER FOR POKING.
PROVIDE SUFFICIENT PASSAGEWAY.
WORKING PLATFORM.
WATER OUTLET
WATER INLET.

FIGS. 2,914 and 2,915.—Elevation and plan of Otto suction gas producer plant. ***It consists of,*** the producer proper F, scrubber J, gas receiver L, and engine O, all connected by piping. ***In operation,*** the piston of the engine moving outwards creates a partial vacuum in the entire gas making apparatus and sucks air into the producer through inlet A. The air passes through the vaporizer B, where it is enriched with water vapor, then through pipes C and D, to the ash pit E. At point C there is an opening through which dry air can be admitted to the mixture going into the ash pit. From the ash pit E, the air is drawn up through the fire F, then through pipe G, three way cock H, and water seal I, to the scrubber J. From the scrubber it passes through pipe K, to bottom of gas receiver L, then out the top through pipe M, to the engine cylinder O. Pipe N, is a vent pipe which is used as an outlet for the gas just before the engine is started. The gas reaching the engine is drawn into the cylinder, where it is mixed with air, which is also drawn in by the suction of the piston. On the next inward stroke of the piston the mixture is compressed and then ignited by an electric system, A, represents the *producer;* B, the *evaporator;* C, the *scrubber;* and D, the *receiver.* The operation of the system may be described as follows:

The gases generated in the producer pass through the evaporator, which is practically a small multitubular boiler, and furnish the sensible heat required for evaporating the water. The resultant vapor is conducted to the ash pit of the producer through the pipe E, by the sucking effect in the producer, while the gas passes from the evaporator to the scrubber filled with coke.

As the gas rises through the interstices of the coke, the washing water descends and not only takes up and removes the dust brought over by the gas, but also clears it of ammonia and other impurities which have a tendency to combine readily with water. From the scrubber the gas passes to the receiver or *suction box.*

The diameter of the receiver being relatively much larger than that of the suction pipe of the engine, the strokes of the engine piston do not therefore cause pulsations between the receiver and the producer.

The producer is usually provided with a charging hopper capable of holding enough fuel for several hours' operation, in the smaller sizes, and allows the admission of fuel to the combustion chamber without permitting access of air thereto during the charging operation.

In operating, a fire is kindled upon the grate, the fuel bed built thereon,

FIGS. 2,914 and 2,915.—*Text continued.*

spark. This burning mixture then expands and forces the piston outward on the power stroke. Again moving inward, the piston forces the burnt gases out into the exhaust vessel P, and then to the atmosphere through pipe Q. ***In charging,*** coal is put in through a charging hopper shown in the illustration at (1). This hopper is closed to the producer while being filled with coal, then is closed to the atmosphere by an air tight door, while the coal is admitted to the fire pot by opening a seal worked lever by (2.) The producer is a heavy steel jacket, provided at the bottom with grates, and lined with fire brick. Resting at the top of the producer is the evaporator B, which is a cast iron water vessel of circular cross section. The use of this water filled evaporator is to keep the top of the producer cool for comfortable working, and to provide the water vapor which is mixed with the air supply, and drawn up through the ash pit and fuel bed. The water vapor is mixed with the air primarily to increase the heat units of the gas by the addition of hydrogen. It also prevents clinkering of the fuel, and keeps the fire cooled down to the proper temperature. The mixture of air and water vapor passes up through the layer of ashes resting directly on the fire grates, into the fire zone where it passes through a bed of red hot fuel. Here the mixture is broken up by the heat, and various chemical reactions take place in different layers of the fire pot above the grates, until finally we have coming out of the producer at G, a gas containing about 25 per cent. of carbon monoxide, and 16 per cent. of hydrogen which is the real producer of gas. This gas has a heating value of approximately 140 heat units per cubic foot. The hot gas before passing out of the producer passes under and around the water contained in the evaporator B, adding to the heat necessary to produce the water vapor. The gas in going through the scrubber, which is merely a steel cylinder partly filled with coke, is thoroughly cleaned from all impurities and foreign substances. Cool water is constantly sprayed down over the coke by the sprinkler shown at (4) in the illustration, and the gas coming up is forced by the coke to follow a crooked passage, bringing it in close contact with the water. When the engine is stopped the gas from the producer is shut off by a three-way valve H, which simultaneously opens the connection to the smoke pipe and the atmosphere. The fire is then kept alive by the natural draught, and the process of gas making can be resumed within a short time by a blower, provided for the purpose of starting or reviving the fire. The pressure in every part of the apparatus, while the engine is running, is always below that of the atmosphere; consequently, no leaks of gas to the outside are possible. A new fire requires about thirty minutes' blowing, while after a shut down over night, ten minutes' blowing is sufficient to produce gas of proper quality for starting the engine.

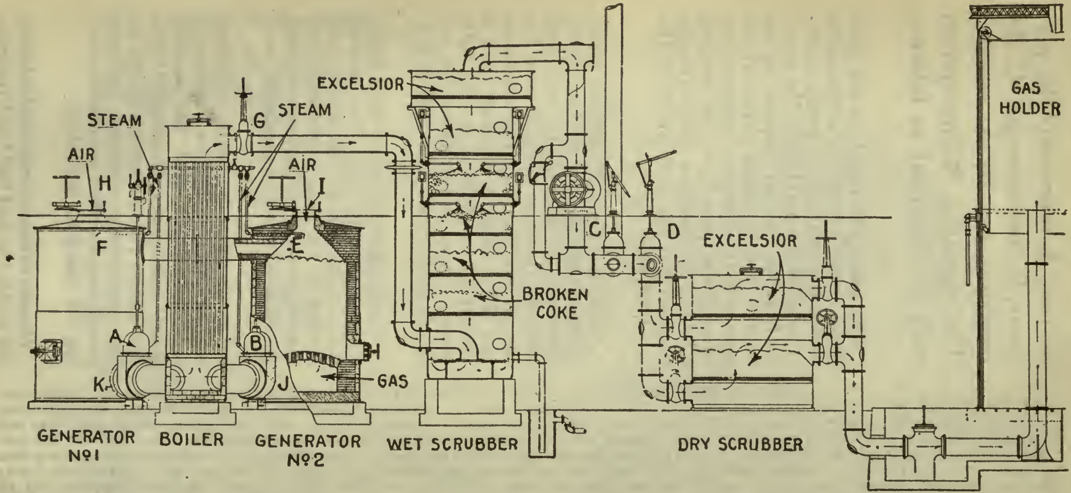


FIG. 2,916.—Sectional view of Loomis-Pettibone suction down draught gas generating system. The unit consists of two generators, No. 1 and No. 2. They are lined with fire brick, including the ash pit and the flues to the boiler, and are equipped with fire brick arches in the bottom, these arches supporting the fuel. Each generator is provided with a door on top for charging fuel and admission of air; also with two cleaning doors above the arches and one, below the arches, opening into the ash pit. Steam connections are made from the boiler to the ash pit and to a point above the fire, in each generator, for use as described under Operation. ***Boiler.*** It is of the multi-tubular type and is connected at the base with the generators by means of brick lined flues. These connections are controlled by water cooled valves A and B. ***The wet scrubber*** consists of a plate steel cylinder provided with trays and water sprays. The trays are charged with coke. The gas inlet in the bottom of the scrubber is submerged in water, and the entering gas bubbles up through the water and, in passing through the trays of coke, meets with the down coming water spray, in this manner being thoroughly washed and cooled. ***The exhauster*** is of the positive type, and is driven by a small steam or gas engine. The exhauster maintains sufficient vacuum on the fires to create a down draught and pressure enough to deliver the gas into the gas holder. The speed of the exhauster is varied to meet the demands for gas. Valves C and D, control the course of the gas delivered by the exhauster. The pipe stock valve C, is only opened when kindling new fires and getting the fires to a proper temperature for gas making, and as soon as this condition is reached, valve C, is closed and D, opened, all the gas being delivered through the dry scrubber to the holder. ***The dry scrubber*** is cylindrical and contains two chambers and trays charged with excelsior or shavings, the chambers being so arranged so that one can be cleaned while the

FIG. 2,916.—*Text continued.*

other is in operation. ***The gas holder*** is of the standard type proportioned to the size of the gas plant. ***In operation,*** fires are kindled with coke and wood in the generators to a depth of about four feet, the exhauster creating a down draught with top doors H, and I, and valves A,B,G, and C, open, and D, closed. As soon as the fires are thoroughly kindled, steam is admitted into the top of the generators at F, and E, and mingles with the air admitted through top doors H, and I, which the operation of the exhauster draws down through the fresh charge of coal and then through the hot fuel bed beneath. The resultant producer or generator gas is drawn down through the grates and ash pits of both generators, valves A, and B, being open, passes up through the vertical boiler, and so on through valve G, to the scrubber and exhauster, valve C, is then closed and valve D, opened, and the gas is driven through the dry scrubber to the gas holder. Coal is charged through the open top doors as needed. The making of gas is according to the demand, as the speed of the exhauster is regulated so as to keep the holder full. The charging doors being open while the air is passing down through them, the operator can see the exact condition of the fires and charge the coal to such parts of the fires as most need it, thereby avoiding the necessity for poking the fires. The condition of the fires is regulated by occasionally passing steam up through one generator and down through the other for a fraction of a minute. This is accomplished by closing the top doors and valve B, and introducing steam at J. The steam is introduced alternately into generators 1 and 2 by using alternately the valves A, and B, and steam inlets J, and K. When bituminous coal, wood, or other fuels containing tar and volatile matters are used in these generators, all the gas and tarry matters distilled from the fresh fuel in the upper strata of the fire pass down through the deep fuel bed, the resultant gas *being fixed and free from tar*. ***Starting.*** From thirty to forty minutes only are required to kindle and get fires into condition for making gas to full capacity of the plant, even when the generators are cold and have not been used for several days. Generators may be stopped for one or two hours and gas making resumed in two minutes. When generators are shut down for six to twenty-four hours, only five to ten minutes are required to resume gas making. ***Wood fuel.*** With this fuel the grates in the generators are covered to a depth of three or four feet with coke, and the exhauster is started with the top doors of generators and valves A,B,G, and C, open and valve D, closed; wood is ignited on top of the coke, and the products from the fires are discharged into the open air through valve C and the stack.

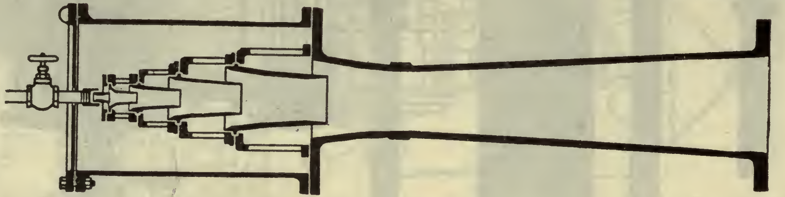


FIG. 2,917.—Amsler producer blower. ***In construction,*** there are a series of nozzles placed on the same axis giving an injector action which enables the steam to move a larger quantity of air against any pressure that is likely to arise in practice. The sliding door at the top permits the regulation of the quantity of air drawn in, while the steam is regulated by the valve. A back pressure valve that can be readily adjusted for pressure will maintain a predetermined pressure on the blower independently of variations in the boiler pressure, and help largely in maintaining constant conditions in the producer.

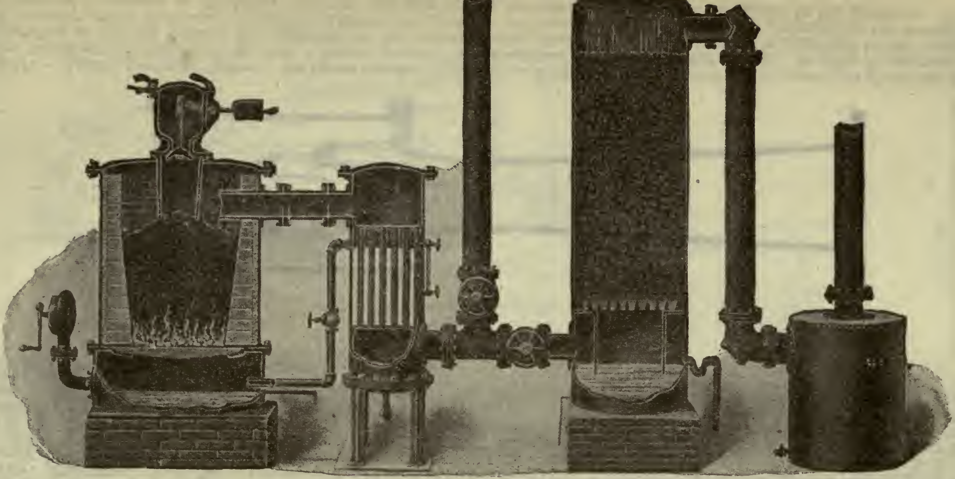


FIG. 2,918.—Sectional view of Backus suction gas producer and engine. The illustration shows (from left to right) 1, generator; 2, vaporizer; 3, scrubber and cooler; 4, gas holder. ***The generator*** consists of a cylinder lined with fire brick. Above the grate this cylinder is filled with fuel. On top of the cylinder is a specially constructed hopper and cover through which the fuel is fed. Attached to the generator is a small blower, which is used when building a new fire, and also just before starting the engine after plant has been idle over night. ***The vaporizer*** is also of cylindrical shape, filled with tubes through which the hot gases pass. Water is admitted around these tubes. While passing through these tubes the hot gas heats the water sufficiently to create steam at atmospheric pressure. The steam or vapor passes to the under side of grate in the producer, adding a certain amount of hydrogen to the gas, thus making a combustible mixture. The ***scrubber*** or ***cooler*** consists of a long boiler steel cylinder having a grate near its lower end. From this grate to a point near the top the scrubber is filled with coke. Through the top water is admitted which sprays over the coke. The gas passing through the coke and spray is freed from impurities and dust which fall to the bottom of scrubber where they can easily be removed through a door put there for that purpose. ***The gas holder*** is simply a steel tank, placed between the scrubber and engine. Any moisture coming with the gas from the scrubber will settle in the holder and can be drawn off as often as may be necessary. ***In operation*** the inhaling action of the engine draws air saturated with steam through the incandescent bed of fuel in the generator, thus forming a combustible gas having an approximate heating value of from 130 to 150 *B.t.u.* per cubic foot. The hot gas, after passing through the vaporizer, enters the scrubber where it is cleaned and cooled and made ready for the engine. The impurities from the gas are drawn off through the bottom of the vaporizer and also from the scrubber by opening a cock or through a door. ***Fuel.*** The fuel best suited is anthracite coal, pea size, although gas coke or charcoal may be used.

and the air necessary for starting combustion supplied by means of a hand or belt-driven *fan*, F. The poor and lean gas produced at starting is first allowed to escape into the open air through the *vent pipe*, G, until the test cock shows that good gas is being produced. The pipe, G, is then closed and the scrubber and receiver are brought into the gas circuit. The engine is now put in operation and as it thereafter performs the function of the fan, the latter is stopped, the operation of the entire system becoming automatic.

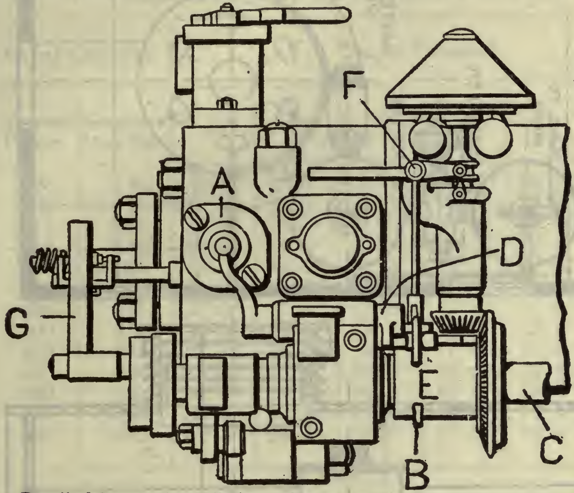
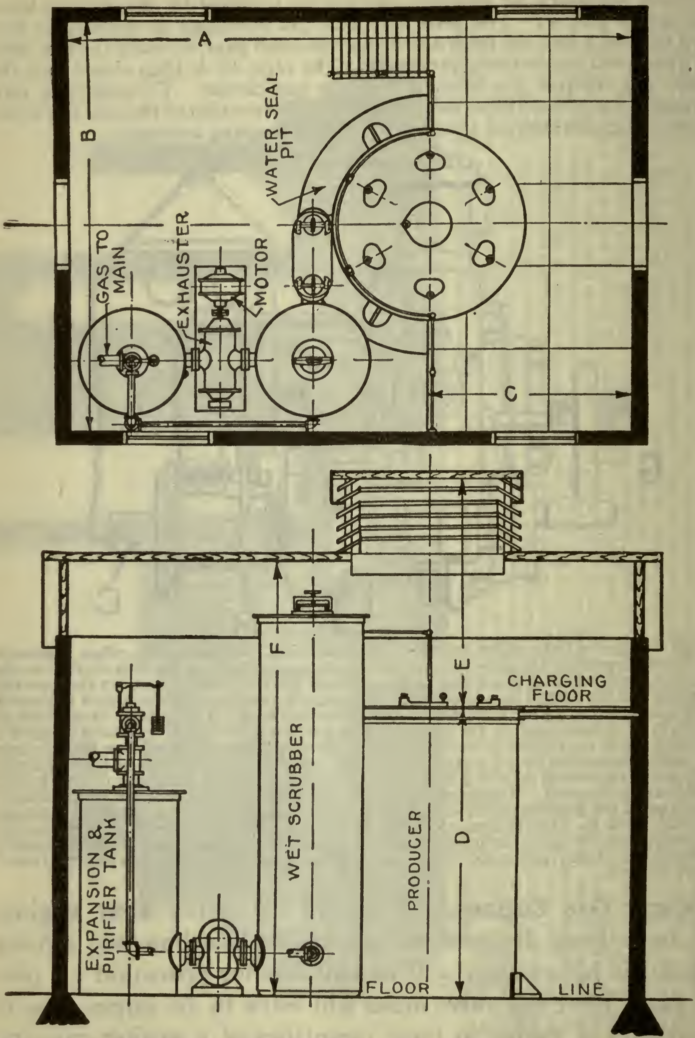


FIG. 2,919.—Detail of Otto engine cylinder, valve gear side. ***In operation,*** when gasoline is used, it is delivered by the gasoline pump driven by an eccentric on the side shaft, into the stand pipe, from which it gravitates to the gasoline valve A, which regulates the quantity of gasoline and the time of its admission, so as to supply automatically the proper mixture of air and gas according to the requirements of the varying load. The valve A, is operated by the cam B, on the side shaft C, the cam actuating a rocker shaft D, the lower arm of which is fitted with a long pin which carries the gasoline roller E. This roller has a free lateral movement which is governed by the vertical arm of the bell crank F, connected with the governor a very slight movement of which is sufficient to place the roller in or out of engagement with the cam on the shaft. The speed of the engine is regulated by the governor which changes the position of the gasoline roller, so that the fuel is admitted only when the speed is normal or slightly below it. The governor is of the fly ball weighted type. Ignition is effected by an electric spark, the igniter being operated by the igniter lever G, actuated by the side shaft. The electrodes of the igniter are tipped with platinum, and the current is derived from a primary battery.

Producer Gas Engines.—It should be noted that engines which have been designed to operate on gasoline fuel require modification before they will be suitable for operation on producer gas. First the valve areas will have to be larger, due to the necessity of taking in large quantities of a weaker gas, and



FIGS. 2,920 and 2,921.—Plan and elevation of Standard Akerlund suction down draught (type V) producer showing space required. The letters refer to the accompanying table.

second, a higher compression will be necessary and advisable for best economy. Compressor pressures according to Durand of from 150 to 180 lbs. should be used.

Rated Capacity of Standard Sizes and Space Required

Akerlund Type B Gas Producers

No.	Rated Capacity		Space Required					
	Horse Power	Coal per hour (lbs)	A	B	C	D	E	F
2	25	40	18′-0″	12′-9″	6′-3″	10′-6″	12′-0″	17′-0″
5	50	75	19′-9″	13′-9″	6′-9″	10′-6″	12′-0″	17′-0″
7	75	115	21′-0″	14′-6″	7′-3″	10′-6″	12′-0″	17′-0″
10	100	150	22′-0″	15′-0″	7′-6″	10′-6″	12′-0″	17′-0″
15	150	225	23′-3″	16′-0″	8′-0″	10′-6″	12′-0″	17′-0″
20	200	300	25′-0″	17′-0″	8′-6″	11′-6″	12′-6″	18′-0″
25	250	375	26′-0″	17′-9″	8′-9″	11′-6″	12′-6″	18′-0″
30	300	450	28′-0″	18′-6″	9′-3″	11′-6″	12′-6″	18′-0″
35	350	525	28′-6″	19′-0″	9′-6″	11′-6″	12′-6″	18′-0″
40	400	600	30′-3″	20′-0″	10′-0″	12′-6″	13′-6″	18′-0″
50	500	750	33′-0″	21′-3″	10′-6″	12′-6″	13′-6″	20′-0″
75	750	1125	36′-9″	22′-9″	11′-6″	14′-0″	14′-6″	22′-0″
100	1000	1500	42′-3″	24′-3″	12′-3″	15′-0″	14′-6″	24′-0″

NOTE.—***Mechanical scrubbers.*** The objection to mechanical scrubbing is that it is not positive. A producer depending upon mechancial apparatus for delivering clean gas ceases to deliver clean gas as soon as the mechanical parts become inefficient. Inefficiency will result from reduction in speed or failure to clean out the apparatus at the proper time. Granting that the mechanical scrubber will deliver clean gas when in perfect condition (which has not been proved in actual practice with all fuels), it is perfectly clear that such apparatus is liable to become ineffective at times due to the slipping of a belt, the burning of a fuse, failure in water supply, clogging of moving parts, or any one of the minor mishaps which may occur in the course of actual operation. Whenever this happens, dirty gas goes over to the engine or to the pipe connections and causes the troubles which the mechanical scrubber is supposed to prevent. As recovered from the mechanical scrubber, tar seldom has a market value as a by-product, and, even in the rare cases where there is a demand for marketable tar at a profitable price, the tar removed from the gas usually has to be treated before it can be sold. Therefore, mechanical scrubbing equipment for bituminous fuel cannot be considered reliable or effective. Moreover, the power required for driving the mechanical scrubber is excessive and this reduces the overall efficiency of the plant.—*Standard Co.*

NOTE.—***Double zone producers.*** Machines of the double zone type, or those which introduce draft from above and from below, must confine themselves to a limited field, because they cannot be recommended for simplicity and ease of operation, and because there are many fuels which will not behave properly in this type of apparatus. There is no way for the operator to tell which zone is doing the work. When the upper zone cools off, or when the fire drops to a lower level, tar goes over with the gas and the producer becomes for all practical purposes a simple updraft producer. The labor of operation is hard, skilled labor is required, and the equipment is both cumbersome and expensive.

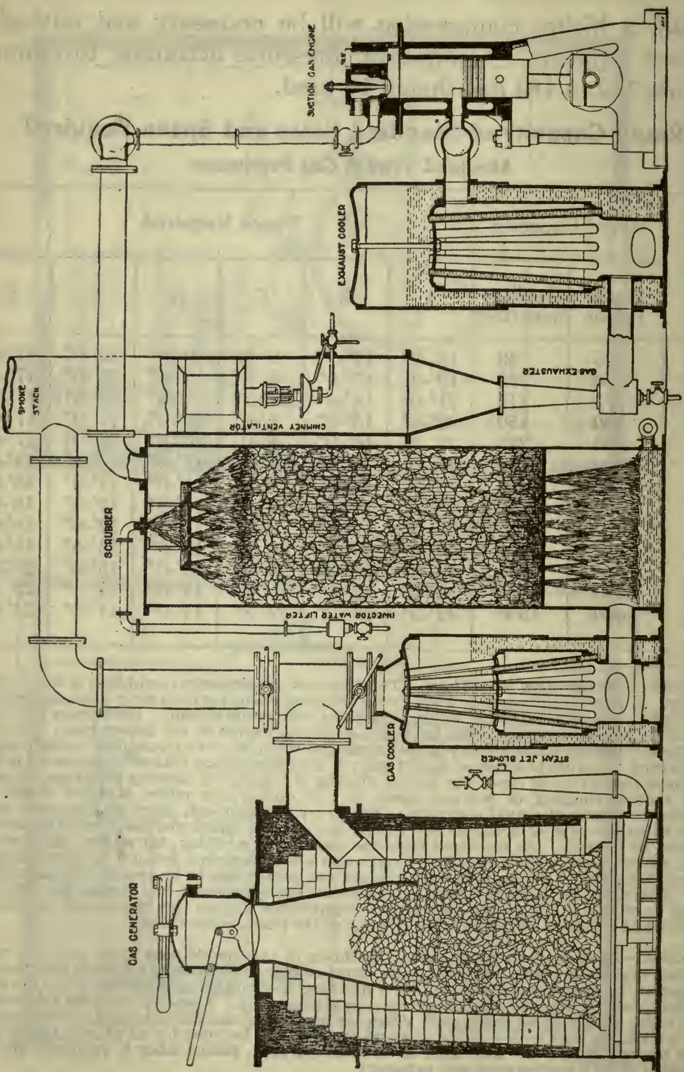


FIG. 2,922.—Diagram of Marine gas producer and engine plant.

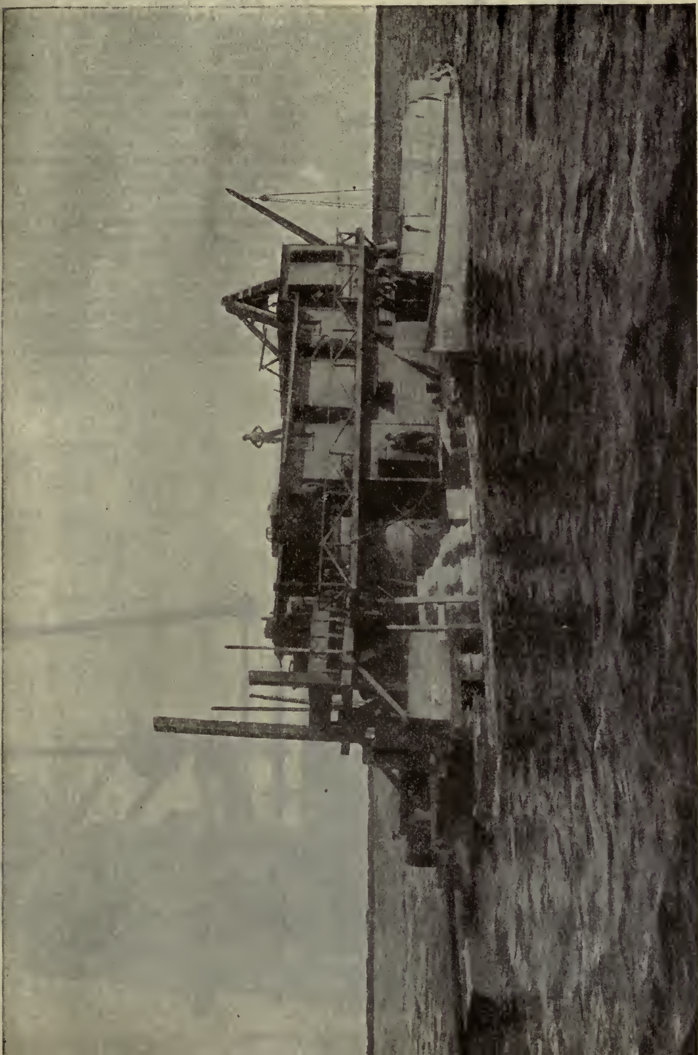


FIG. 2,923.—A 200 horse power Standard (Akerlund type B) suction down draught producer operating a suction dredge. ***The illustration shows*** the dredge working in deep salt water. Salt water is used in the scrubber. Full load operating, 24 hours per day is essential.

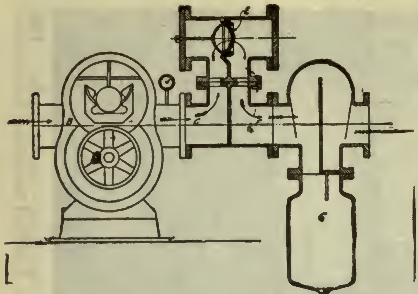


FIG. 2,924.—Smith type F, tar extractor. ***In operation,*** the producer gas partially cooled in the primary condenser, enters the rotary exhauster B, which delivers it under two or three pounds pressure to the gas main C. The gas then passes through the mat of glass wool E, at high velocity and with a drop in pressure of one-half to two pounds. The raw gas entering the mat of glass wool contains the tar in the form of a fog, there being millions of particles of the tar per cubic inch of the gas. In its passage through the mat of glass wool a peculiar change occurs, the particles in the tar fog becoming agglomerated into large drops. These fall out of the gas by gravity into the trap G, and the cleaned gas passes to the distributing system. The tar extractor is in no sense a filter. The tar is removed from the gas and collected as described above, no deposited tar remaining in the mat of glass wool which stays serviceable continuously. The advantages over mechanical scrubbing are greater cleanness, less power and water required and easier handling of tar because of less entrained water.

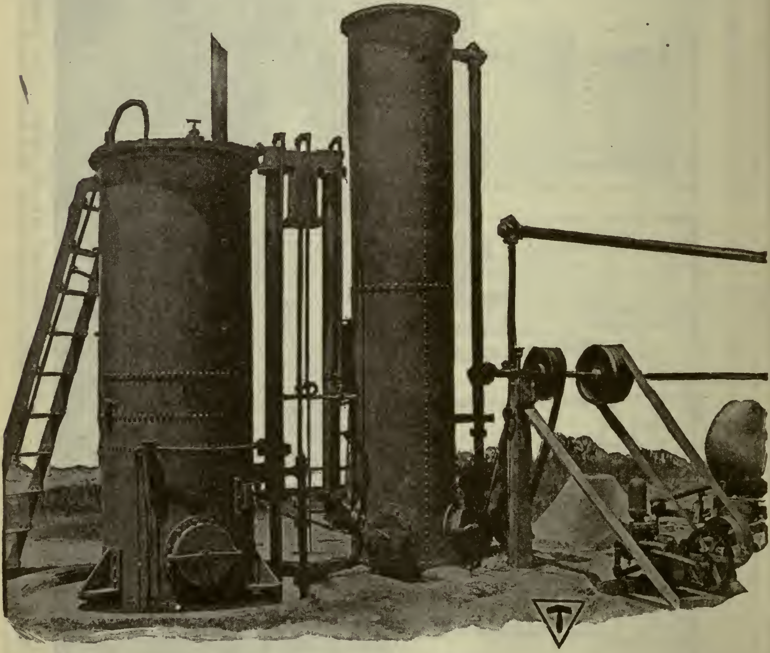
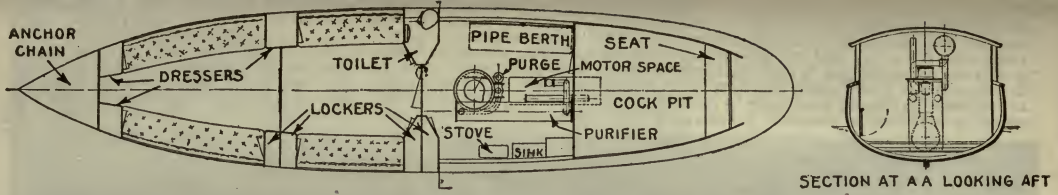


FIG. 2,925.—Akerlund small size suction down draught producer.



FIGS. 2,926 and 2,927.—Plan and sectional view of a cruiser showing installation of marine producer plant and engine.

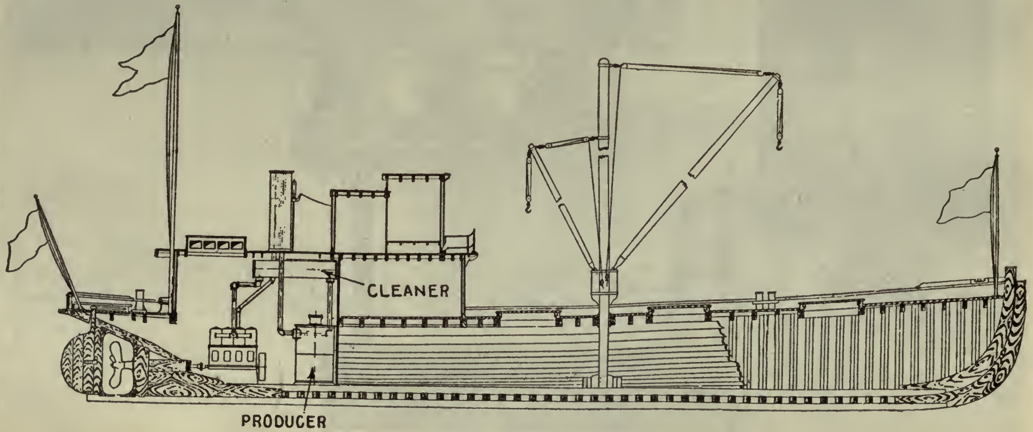


FIG. 2,928.—Sectional view of 115′×24′ freight boat with 150 *h.p.* producer plant showing general arrangement of the apparatus and low center of gravity.

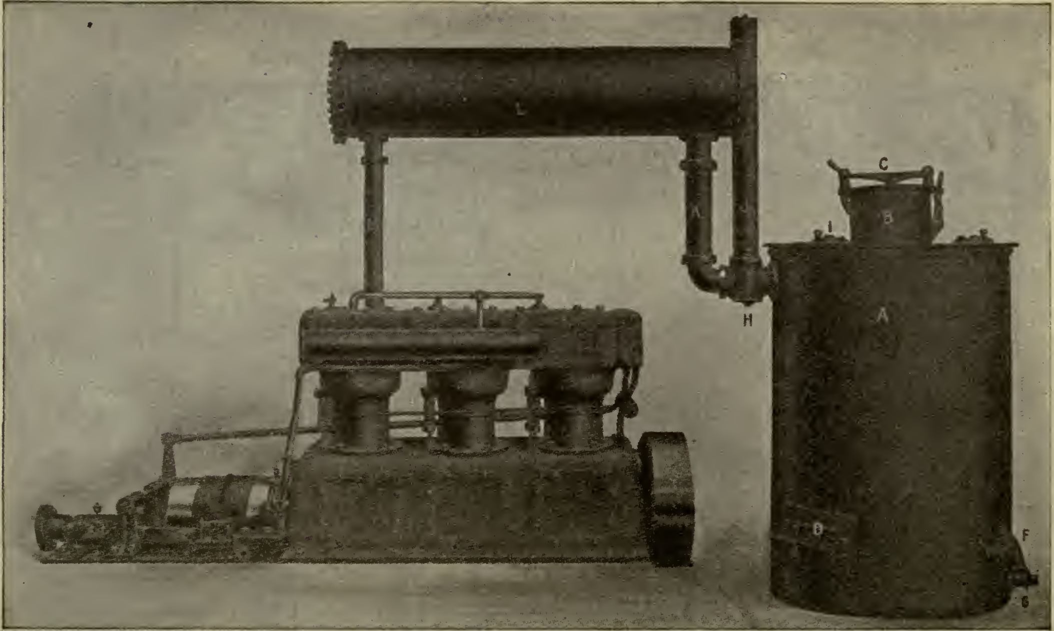


FIG. 2,929.—Marine Producer Gas Power Co. producer plant and engine. ***The parts are*** A, producer or generator; B, hopper through which coal is fed through air tight seal to producer; C, latch; D, cleaning door; F, small hand blower; G, unsealed port holes; H, vent valve. ***In operation,*** when blowing the fire up or starting a new fire, H, is opened to the pipe J, which is the vent pipe out to the open air. When the fire is in proper condition, the valve H, is closed to the pipe J, and opened to the pipe K, which leads into the gas cleaner L, and through this cleaner the gas is driven down through the pipe M, or the inlet pipe to the engine, into the engine cylinders. The engine then is operated in the same way as a gas or gasolene engine would be operated. The cleaner L, is a cylindrical tank into which the cleaning water flows, coming from the pump on the engine, to cool the gas. This water is then allowed to flow overboard from the cleaner.

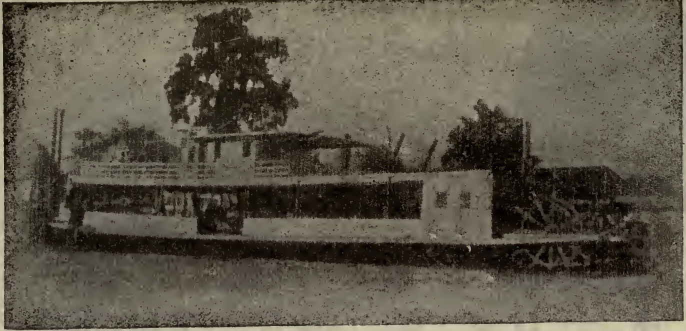


FIG. 2,930.—Light draught freight boat. Exterior view showing general appearance of the boat.

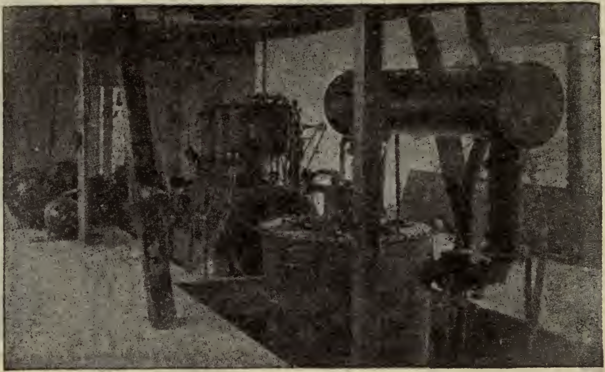
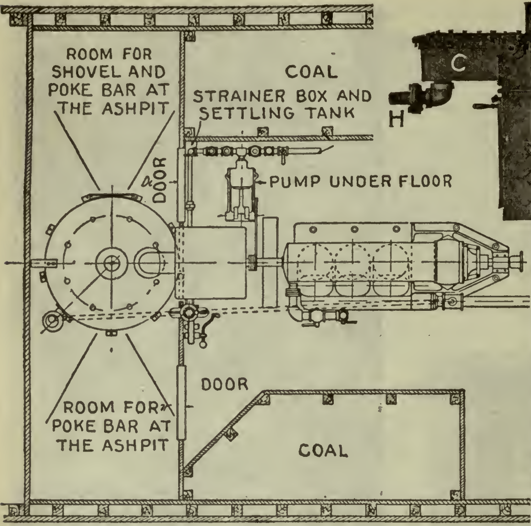


FIG. 2,931.—Light draught freight boat Interior view showing power plant. ***The boat*** is 114 ft. 6 in. overall length, 23 ft. 2 in. beam, and 3 ft draught. Equipped with 75 *h.p.* Marine Producer Gas Power Co. producer Speed 8 miles per hour.


ROOM FOR SHOVEL AND POKE BAR AT THE ASHPIT
COAL
STRAINER BOX AND SETTLING TANK
DOOR
PUMP UNDER FLOOR
DOOR
ROOM FOR POKE BAR AT THE ASHPIT
COAL

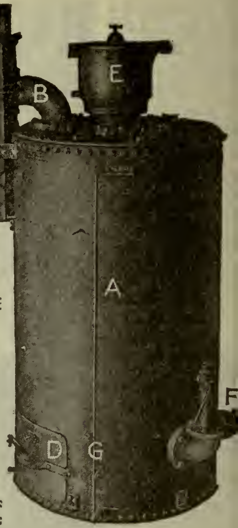
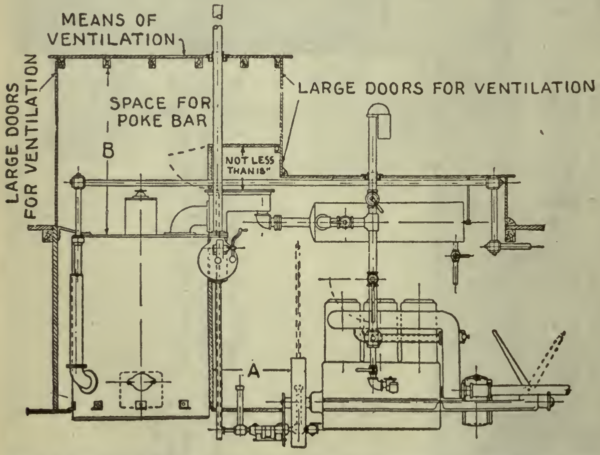


FIG. 2,934.—Nelson producer plant. **A**, producer; **B**, gas elbow connects producer with, **C**, scrubber and drier; cleaning door for removing ashes; **E**, hopper with air seal; **F**, an inlet for the steam and air entering the fire box; **G**, grate shaker; **H**, gas outlet to engine.



FIGS. 2,932 and 2,933.—Plan and elevation of typical Galusha marine producer plant installed in boat.

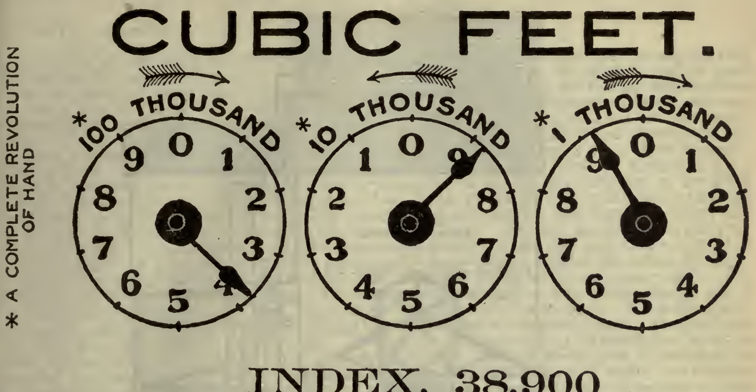
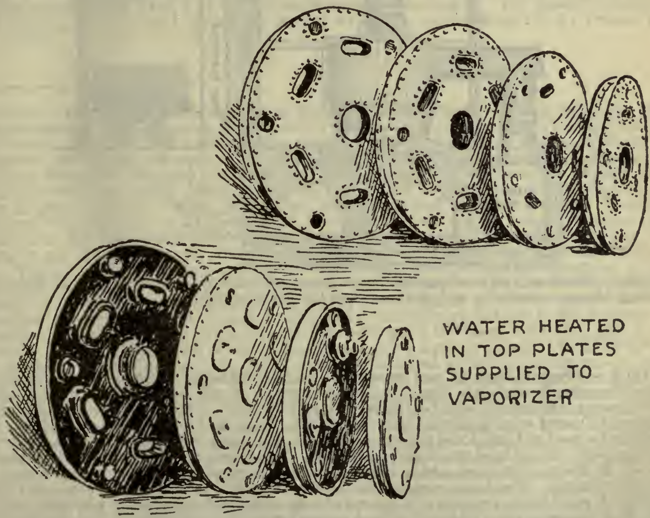


FIG. 2,937.—How to read a gas meter. Each hand moves in a different direction as indicated by the arrows. Read the figures *that the hands have actually passed*, beginning with the dial at the left, and add two ciphers to the right of the reading. Subtract the last month's reading from the above index and the difference will be the gas used to date.



FIGS. 2,935 and 2,936.—Akerlund, type B water cooled top plates, showing construction.

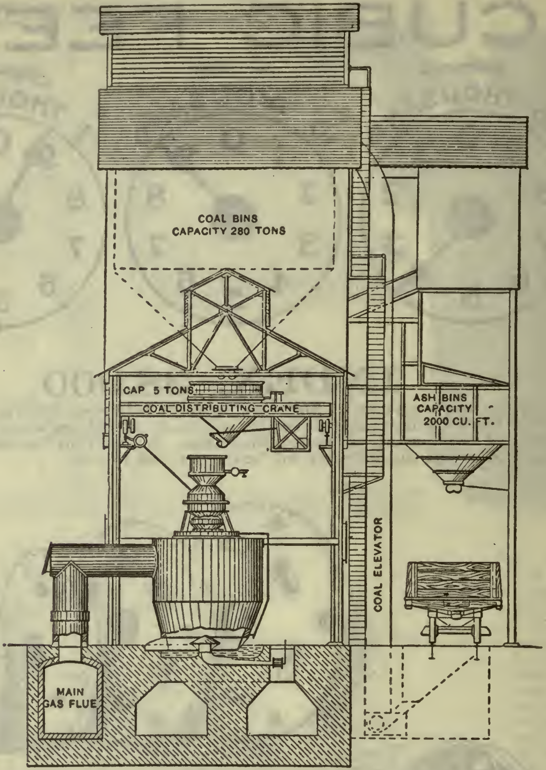


FIG. 2,938.—Sectional view of gas producer house at Lackawanna Steel Co., Buffalo, N. Y., showing Morgan producer and coal and ash handling equipment.

NOTE.—***Proportions of gas producers and scrubbers.***—Small inside diameter means excessive draught through the fire. If a fire be forced, as will be necessary with too small an inside diameter, the results will be clinkers and blow holes or chimneys through the fire bed, with excess CO_2 and weak gas; clinkers fused to the lining, and burning out of grates. If sufficient steam be used to keep down the excessive heat, the result is likely to be too much hydrogen in the gas, with the attendant engine troubles. The lining should never be less than 9 in. thick even in the smaller sizes, and a 100 *h.p.*, or larger, producer should have at least 12 in. of generator lining. The lining next to the fire bed should be of the best quality of refractory material. A good lining consists of a course of soft common bricks put in edgewise next to the steel shell of the generator, laid in Portland cement; then a good fire brick 6 in. thick laid inside to fit the circle, the bricks being dipped as laid in a fine grouting of ground fire brick. If we

NOTE.—*Continued.*

take 1¼ lbs. of coal per *h.p.*-hour as a fair average and 10 lbs. of coal per hour per square foot of internal fuel bed cross section, with 9 in. of refractory lining up to 100 *h.p.* and at least 12 in. of lining on larger sizes, the generator will give good gas without forcing and without excessive heat in the zone of complete combustion. A 200 *h.p.* producer on this basis consumes 250 lbs. of coal at full load, and at 10 lbs. per sq. ft. internal area 25 sq. ft. will be necessary. With a 12 in. lining, the outside diameter will be 92 in. Practice has shown that the depth of the fuel bed should never be less than the inside diameter up to 6 ft.; above this size, the depth can be adjusted as experience indicates the best working results. Assuming for a 200 *h.p.* producer 18 in. for the ash pit below the grate, 12 in. for the thickness of the grate and the ashes to protect it, 68 in. depth of fuel bed, 24 in. above the fuel to the gas outlet, the height will be 10 ft. 4 in. to the top of the generator; above this the coal feeding hopper, say 32 in. high, is mounted; this makes the height over all 13 ft. The wet scrubber of a gas producer should be of ample size to cool the gas to atmospheric temperature and wash out most of the impurities. A good rule is to make its diameter three-fourths that of the inside diameter of the generator and the height one and one-half times the height of the generator shell. For a 100 *h.p.* producer, 4 ft. inside diam., the wet scrubber should be 3 ft. inside diam., and if the generator shell be 8 ft. 6 in. high, the scrubber should be 12 ft. 9 in. high. When filled with the proper amount of baffling and scrubbing material (coke is commonly used), the scrubber will have space for about 30 cu. ft. of gas. A 100 *h.p.* gas engine using 12,000 *B.t.u.* per *h.p.-hour* will use 160 cu. ft. of 125 *B.t.u.* gas per minute. The wet scrubber will therefore be emptied 5⅓ times every minute, and would require about 8⅓ gallons of water per minute; if the diameter of the scrubber were reduced one-third, the volume of water necessary to cool and scrub the gas would have to be doubled. Gas must be cooled below 90° F. to enable it to give up the impurities it carries in suspension, and even lower than this to condense its moisture. A separate dry scrubber with two compartments should always be provided and the piping between the two scrubbers so arranged that the gas can be turned into either part of the dry scrubber at will. The dry scrubber should be equal in area to the inside of the generator and the depth of each part should be sufficient to accommodate at least 2 cu. ft. of scrubbing material and give 1 cu. ft. of space next to the outlet. Oil soaked excelsior is a good scrubbing material and should be packed as closely as possible.—*F. C. Tryon, Power, Dec. 1, 1908.*

NOTE.—***The best coal to buy*** for a producer in any locality is that which by analysis or calorimeter test shows the most heat units for a dollar. It rarely pays to buy gas coal unless it can be had at a moderate cost over the ordinary steam bituminous grade. For very high temperature melting operations a fairly high percentage of volatile matter is necessary to give a luminous flame and intensify the radiation from the roof of the furnace. Freely burning gas coals are the most easily gasified, and the capacity of the producer to handle these coals is twice as great as when a slaty, dirty coal, high in ash and sulphur, is used. It is usually best to use "run-of-mine" coal, crushed at the mine to pass a 4-in. ring. It never pays to use slack coal, for it cuts down the capacity by choking the blast, which has to be run at high pressure to get through the fire, overheating the gas and lowering the efficiency of the producer. There is always a certain amount of CO_2 formed, even in the best practice; in fact, it is inevitable, and if kept within proper limits does not constitute a net loss of efficiency, especially with very short gas flues, because the energy of the fuel so burned is represented in the sensible heat or temperature of the gas, and results in delivering a hot gas to the furnace. The best result is at about 4% CO_2, a gas temperature between 1,100° and 1,200° F., and flues less than 100 ft. long. The amount of steam required to blow a gas producer is from 33% to 40% of the weight of the fuel gasified. If 30 lbs. of steam be called a standard horse power, we have therefore to provide about 1 *h.p.* of steam for every 80 lbs. of coal gasified per hour or for every ton of coal gasified in 24 hours. In the original Siemens air blown producer about 70% of the whole gas was inert and 30% combustible. Then with the advent of steam blown producers the dilution was reduced to about 60%, with 40% combustible. Now, under the system of automatic feed, uniform conditions, perfect distribution and adjustment of the steam blast here presented, we are able to reduce the nitrogen to 50% and sometimes less. In the best practice the volume of gas from the producer is now reduced to about 60 cu. ft. per pound of coal, of which 30 cu. ft. are nitrogen. These volumes are measured at 60° F. The temperature of the gas leaving the producer under best modern conditions is about 1,200° F. It can be run cooler than this, but not much, except at a sacrifice of both quantity and quality. At this temperature, the sensible heat carried by the gas is $1,200 \times .35$ (average specific heat) $= 420$ *B.t.u.* per pound. As one pound of good gas is about 16 cu. ft. and carries about 16×180

NOTE.—*Continued.*

=2,880 heat units at normal temperature, we see that the sensible heat carried away represents about one-seventh, or over 14% of the combustive energy, which is much too large a percentage to lose whenever it can be utilized by using the gas at the temperature at which it is made. ***Capacity of producers.***—The capacity of a gas producer is a varying quantity dependent upon the construction of the producer and upon the quality of the coal supplied to it. The point is, not to push the producer so hard as to burn up the gas within it; also to avoid blowing dust through into the flues. These two limitations in a well constructed automatically fed gas producer occur at about the same rate of gasification, namely, at about 10 lbs. per sq. ft. of surface per hour with bituminous coal carrying 10% of ash and 1½% of sulphur. With gas coal, having high volatile percentage and low ash, this rate can be safely increased to 12 lbs. and in some cases to 15 lbs. per sq. ft. At 10 lbs. per sq. ft. the capacity of a gas producer 8 ft. internal diameter is 500 lbs. per hour, which with gas coals may be increased to a maximum of about 700 lbs. It frequently happens that the cheapest coal available is of such quality that neither of these figures can be reached, and the gasification per sq. ft. has to be cut down to 6 or 7 lbs. per hour to get the best results. ***Flues.***—It is necessary to provide large flue capacity and to carry the full area right up to the furnace ports, which latter may be slightly reduced to give the gas a forward impetus. Generally speaking, the net area of a flue should not be less than 1/16 of the area of the gas making surface in the producers supplying it. Or it may be stated thus: The carrying capacity of a hot gas flue is equivalent to 200 lbs. of coal per hour per sq. ft. of section. ***Loss of energy in a gas producer.***—The total loss from all sources in the gasification of fuel in a gas producer under fairly good conditions, when the gas is used cold or when its sensible heat is not utilized, ranged between 20% and 25%, which under very bad conditions may be increased to 50%. The loss under favorable conditions, using the gas hot, is reduced to as low as 10%, which also includes the heat of the steam used in blowing.—*Morgan Construction Co.*

CHAPTER 50

AEROPLANES

An aeroplane may be defined as a *heavier than air flying machine;* it consists essentially of:

1. A body or frame (ill advisedly called fuselage or nacelle*) which carries
2. The power plant, passengers, etc., and to which is attached
3. The landing gear, or under carriage,
4. The wings (planes) and
5. Control devices.

In order to understand the text, the following definitions of terms used in connection with aeroplanes should be understood:

Definitions

Aerofoil—A lifting surface or wing.

Ailerons—Movable hinged surfaces for control of *rolling* or *banking*.

Air pocket—A local movement of the air causing an aeroplane to drop.

Angle of incidence—The angle at which a wing is inclined to the line of flight.

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*NOTE.—The practice of introducing foreign terms in place of plain English cannot be too strongly condemned. There is no sane reason for such practice, in fact it reflects an egotistical desire to display a knowledge(?) of some other language.

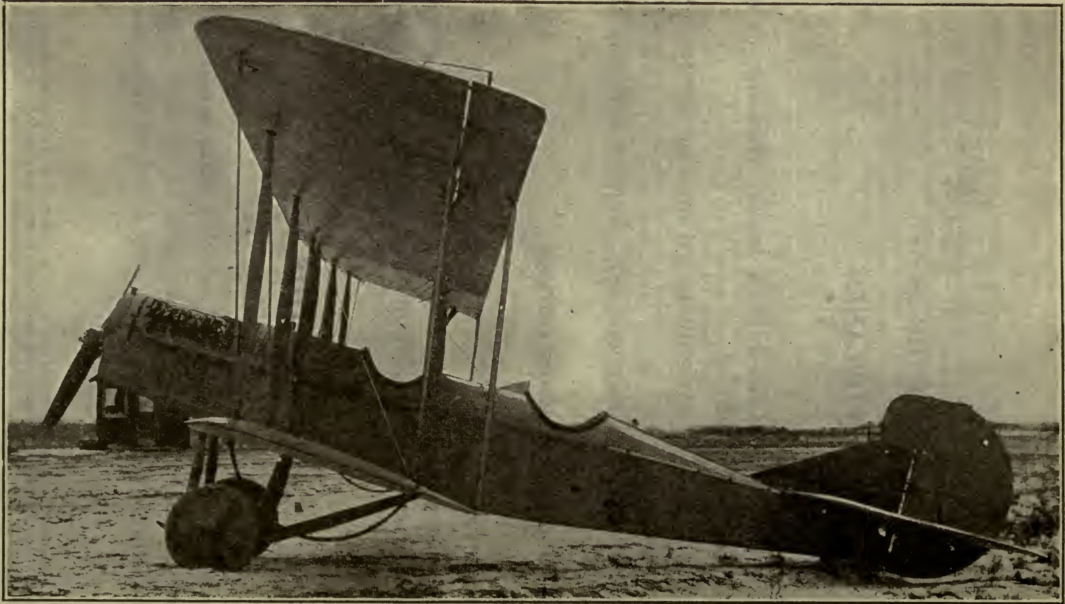


FIG. 2,939.—Curtiss biplane tractor. It has a maximum horizontal speed of 75 miles per hour and a minimum of 43 miles per hour, and will climb 3,000 ft. in ten minutes. Powered with an 8-cylinder, "V" type Curtiss "OX" engine, which develops 90 horse power at 1,400 *r.p.m.*; it consumes slightly more than one-half pound of gasoline per horse power per hour. With a wing span of 43 feet, 7⅔ inches and an overall length of 27 feet 3 inches, it weighs empty only 1,405 pounds but will carry a useful load of 485 pounds. It is equipped with dual control, carries two passengers "tandem," each in a separate cockpit.

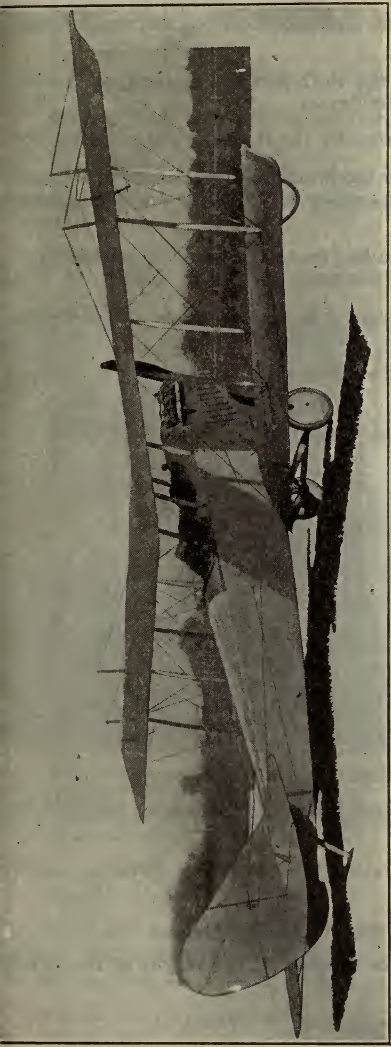


FIG. 2,940.—Curtiss biplane tractor as described in fig. 2,939, view from rear.

Aspect ratio—The long span of the wing surface divided by the width.

Bank—To tilt sideways in turning.

Body—The main frame of an aeroplane which covers the power planes and operator, and to which are attached the wings, landing gear, etc.

Body skid—Flexible support usually placed in front of the landing gear.

Camber—The depth of the curve given to the surface of a wing; curvature.

Chord—Distance between the entering edge and trailing edge of a wing measured on a straight line touching front and rear bottom point of a wing.

Cockpit—The opening and space in the body where the passengers sit.

Control braces—Steel struts on the controls to which the control wires are attached.

Controls—Devices used for operating the adjustable surfaces and for regulating the movements of the machine in flight.

Critical angle—The steepest angle that can be assumed by an aeroplane for a given speed without loss of control.

Decalage—Difference in angle of incidence between any two distinct aerofoils.

Dope—Preparation for treating the cloth surfaces to produce tautness and to maintain air and moisture tightness.

Drag—The total resilience overcome by the thrust of the propeller.

Drift—An objectionable term for the horizontal component of the force due to the air striking the inclined surface of the wings and which tends to push the machine backward.

Drift wires—Horizontal wires leading from the nose of the frame to the wings to prevent collapsing toward the rear.

Elevator—A hinged horizontal control surface acting to lift or depress the tail of the machine.

Pairing—Building up a member or part with a false piece to give it a stream line.

Fineness ratio—The fore and aft length of the body divided by the greatest width across the wind.

Flaps—Hinged horizontal control surfaces cut out of the wings.

Flying wires—Those supporting the body when in flight.

Fuselage—A term sometimes used for body.

Gap—The distance between the upper and lower wings of a biplane.

Gliding angle—The angle of inclination assumed by an aeroplane when descending with power shut off.

Hangar—An aeroplane shed.

Horizontal fin—The horizontal fixed tail plane.

Horizontal stabilizer—The horizontal fixed tail plane.

Horns—Steel struts on the controls to which the control wires are attached.

Joy stick—A control lever operating the controlling surfaces, usually the wing flaps and elevator.

Landing gear—The wheels and supports for landing.

Landing wires—The single wires supporting the weight of the panels when landing or on the ground.

Leading edge—The front or entering edge of a plane.

Lift—The vertical component of the pressure of the air acting on the inclined surfaces of the wings and which tend to lift the machine.

Longerons—The fore and aft members of the body framework.

Longitudinals—The fore and aft members of the body framework.

Monoplane—An aeroplane having only one lifting surface or pair of wings.

Nose dive—A dangerously steep descent.

Outrigger—The framework connecting the main surface with an elevator placed in advance of it.

Pancake—To stall.

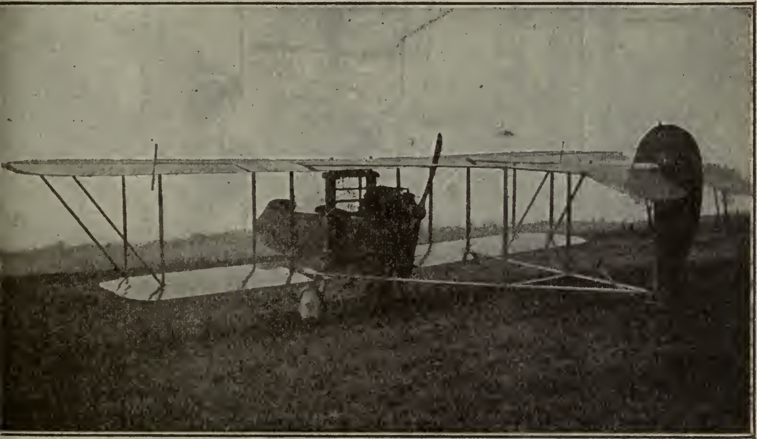


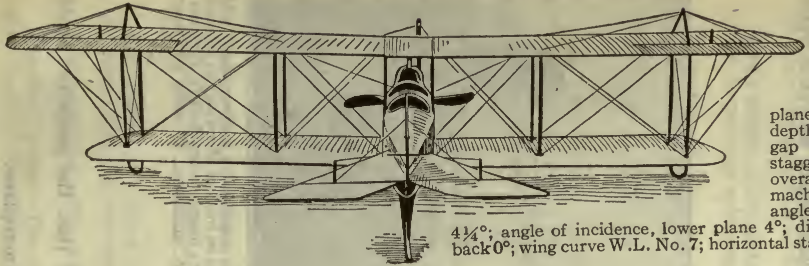
FIG. 2,941.—Burgess pusher observation biplane. A machine designed for medium speed with great weight carrying capacity. The observer is extended in front of the wings to increase the range of vision. ***General specifications:*** span 45 ft.; length over all 31 ft. 6 inches; horse power, 140; live load, 920 lbs.; speed, maximum, 85, minimum, 47 miles per hour; seating capacity, 2; fuel, enough for 4 hours.

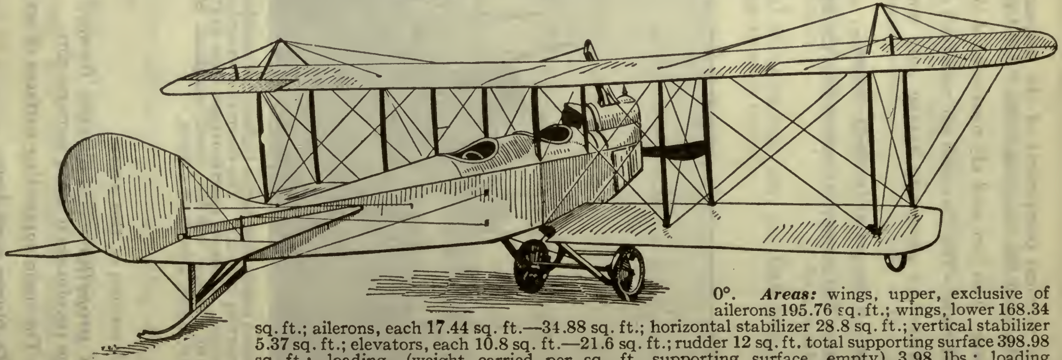
Panel—A section of the wings.

Pitch of propeller—The distance forward that the propeller would travel in one revolution if there were no slip.

Planes—The main supporting surfaces of an aeroplane.

Pitch—To plunge nose down.





FIGS. 2,942 and 2,943. —Wittemann - Lewis model TT aeroplane. Specifications. ***General:*** wing span, upper plane 42 ft.; lower plane 34 ft. ***Dimensions:*** depth of wing chord 5 ft. 6 in.; gap between wings 5 ft. 6 in.; stagger 0°; length of machine overall 27 ft. 2 in.; height of machine overall 11 ft. 6½ in.; angle of incidence, upper plane 4¼°; angle of incidence, lower plane 4°; dihedral angle 1½°; sweepback 0°; wing curve W.L. No. 7; horizontal stabilizers, angle of incidence 0°. ***Areas:*** wings, upper, exclusive of ailerons 195.76 sq. ft.; wings, lower 168.34 sq. ft.; ailerons, each 17.44 sq. ft.—34.88 sq. ft.; horizontal stabilizer 28.8 sq. ft.; vertical stabilizer 5.37 sq. ft.; elevators, each 10.8 sq. ft.—21.6 sq. ft.; rudder 12 sq. ft. total supporting surface 398.98 sq. ft.; loading, (weight carried per sq. ft. supporting surface, empty) 3.98 lbs.; loading (weight carried per sq. ft. supporting surface, loaded) 5.41 lbs.; loading per B. H. P. loaded 21.6 lbs. ***Weights:*** net weight machine empty 1560 lbs.; gross weight machine and load 2165 lbs.; useful load 605 lbs.; fuel 220 lbs.; oil 35 lbs.; pilot 175 lbs.; passenger or other 175 lbs.; total 605 lbs. ***Performance:*** speed, maximum, horizontal flight 74 miles per hr.; speed, minimum 38 miles per hr.; climbing speed 3,000 ft. 10 min. ***Motor:*** Hall-Scott—4 cylinder, water cooled 100 H.P.; weight per rated H.P. 4.45 lbs.; bore and stroke 5¼″ × 7″; fuel consumption per hour 10 gallons; fuel tank capacity 36 gallons, oil capacity provided in crank case 5 gallons. ***Propeller:*** material, wood; pitch, according to requirements of performance, diameter, according to requirements of performance; direction of rotation, as viewed from pilot's seat, clockwise. ***Covers:*** canvas engine hood, propeller and cockpit covers for protection when exposed to the weather, is furnished with each machine.

Pusher—A rear drive aeroplane; one that is *pushed* instead of *pulled* through the air.

Pylon—Any V-shaped construction from the point of which wires are taken.

Rib—Framework members transverse to the spars.

Roll—To turn about the longitudinal axis.

Rudder—A hinged surface giving lateral control, the same as a ship's rudder.

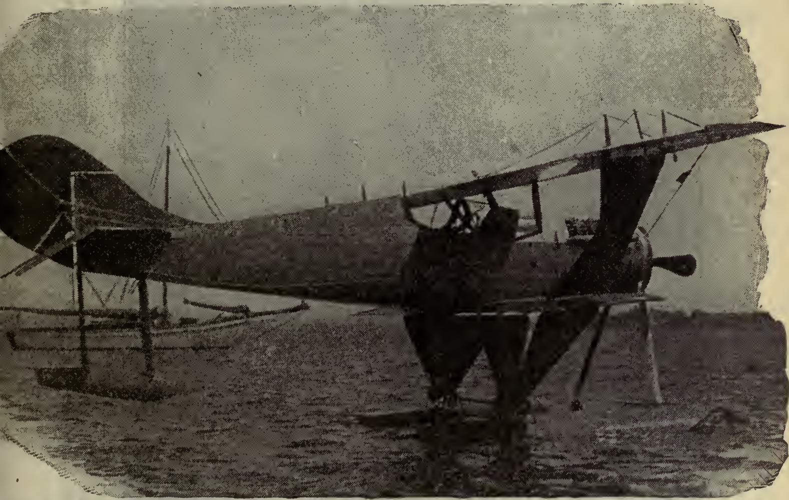


FIG. 2,944.—Burgess-Dunne flying boat. It is designed primarily for sportsmen who do not wish to lose the sensation of the speed boat, but who also wish to broaden their scope of pleasure by flying. ***General specifications:*** span, 53 ft.; length all over 25 ft. 2 inches; horse power, 100; live load, 560 lbs.; speed, maximum, 68, minimum, 43 miles per hour; seating capacity, 3; fuel enough for 3 hours.

Side slip—To fall due to excessive bank or roll.

Skid—To be carried sideways by centrifugal force when turning to right or left.

Slip—The difference between the actual forward travel of a propeller in one revolution and its pitch.

Spread—The distance from tip to tip of the wings.

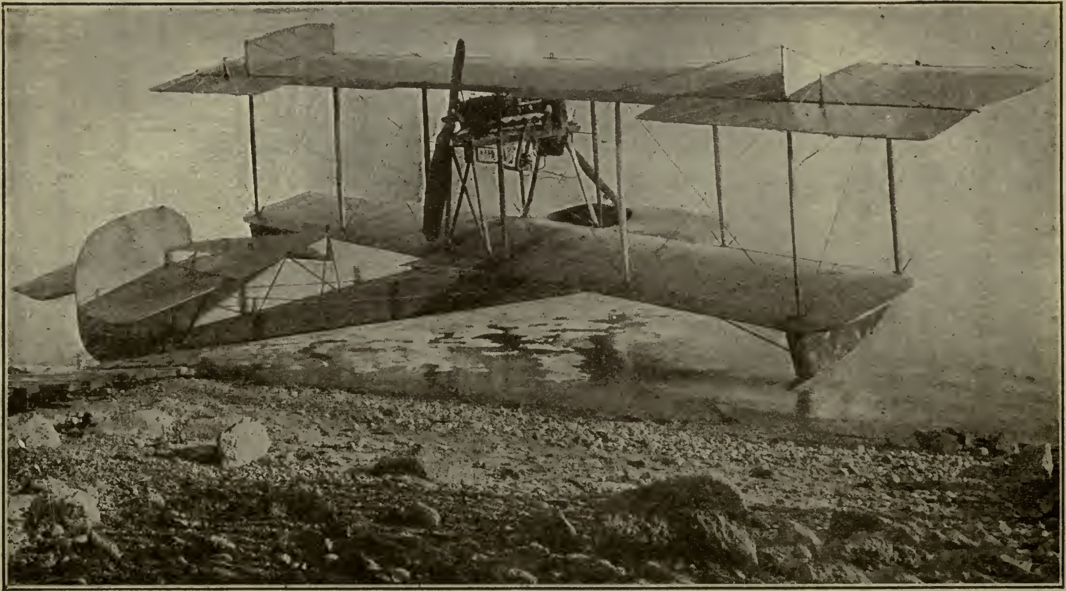


FIG. 2,945.—Curtiss model F, flying boat; side view. It is of the pusher type. Speeds: minimum 45 miles per hour; maximum 65 miles per hour. It will climb at the rate of 1,500 feet in 10 minutes. Capacity two passengers.

Stability—The property of an aeroplane to maintain its direction and return easily to its equilibrium against disturbing conditions.

Stabilizer—A surface, such as a fin or tail plane designed to give inherent stability.

Stagger—The horizontal distance that the entering edge of the upper wing is ahead of the entering edge of the lower wing.

Stalling—Loss of control in assuming angles of incident greater than the control angle.

Stream line—The shape of a body or part which presents the least resistance to the wind.

Struts—Members used to take pressure or compression.

Tail—The rear surfaces of the machine.

Tail skid—Flexible support under the tail.

Toss—To plunge tail down.

Tractor—A front drive aeroplane; one that is *pulled* instead of *pushed* through the air.

Trailing edge—The rear edge of a plane.

Undercarriage—The wheels and supports for landing.

Vertical fin—The small vertical fixed plane in front of the rudder.

Vertical stabilizer—The small vertical fixed plane in front of the rudder.

Volplane—A gliding descent.

Wash—The disturbance of air produced by the flight of an aeroplane.

Wash in—An increasing angle of incidence of a wing surface toward its tip.

Wash out—A decreasing angle of incidence of a wing surface toward its tip.

Warp—To distort a controlling surface.

Wing bars—The longitudinal spars of the interior wing framework.

Wings—The main supporting surfaces of an aeroplane.

Wing skid—Flexibility support under the lower wing.

Wing tip—The right or left hand extremity of a surface.

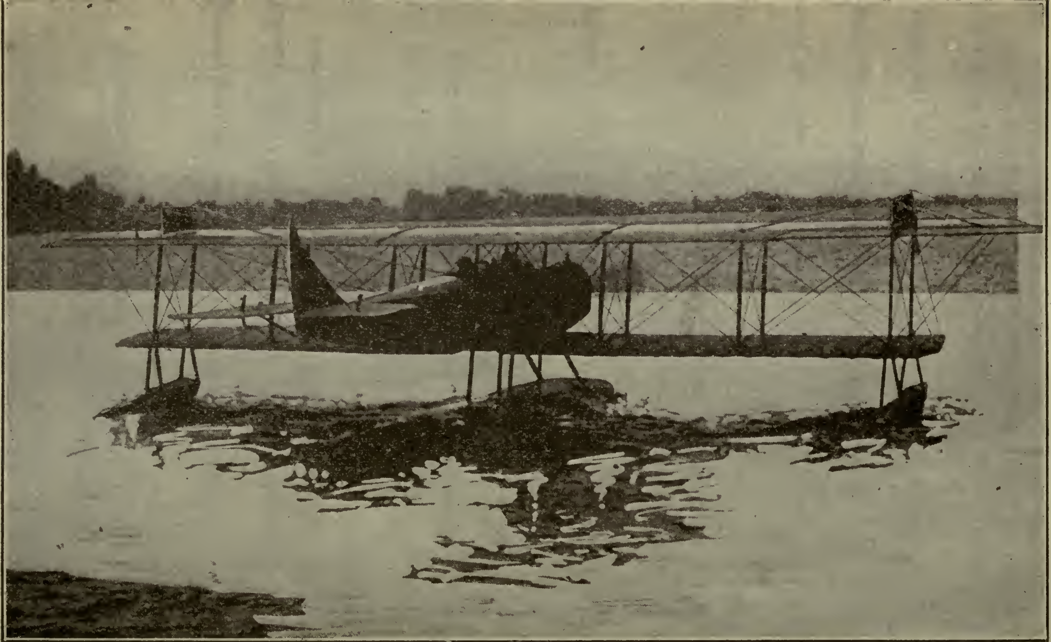
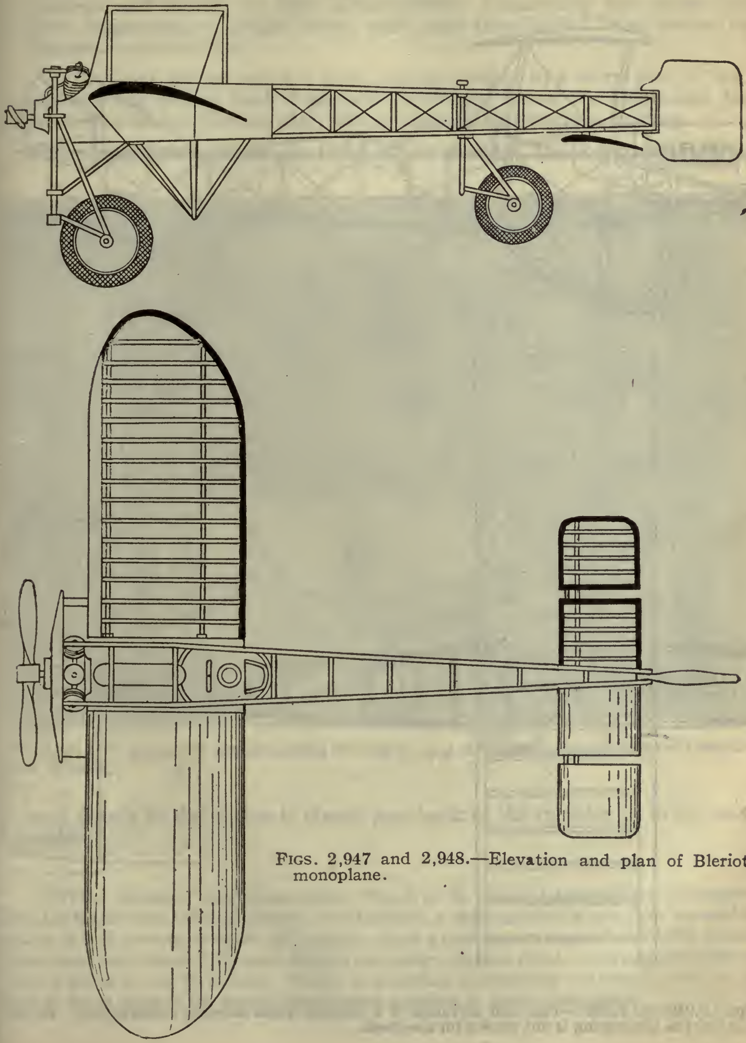
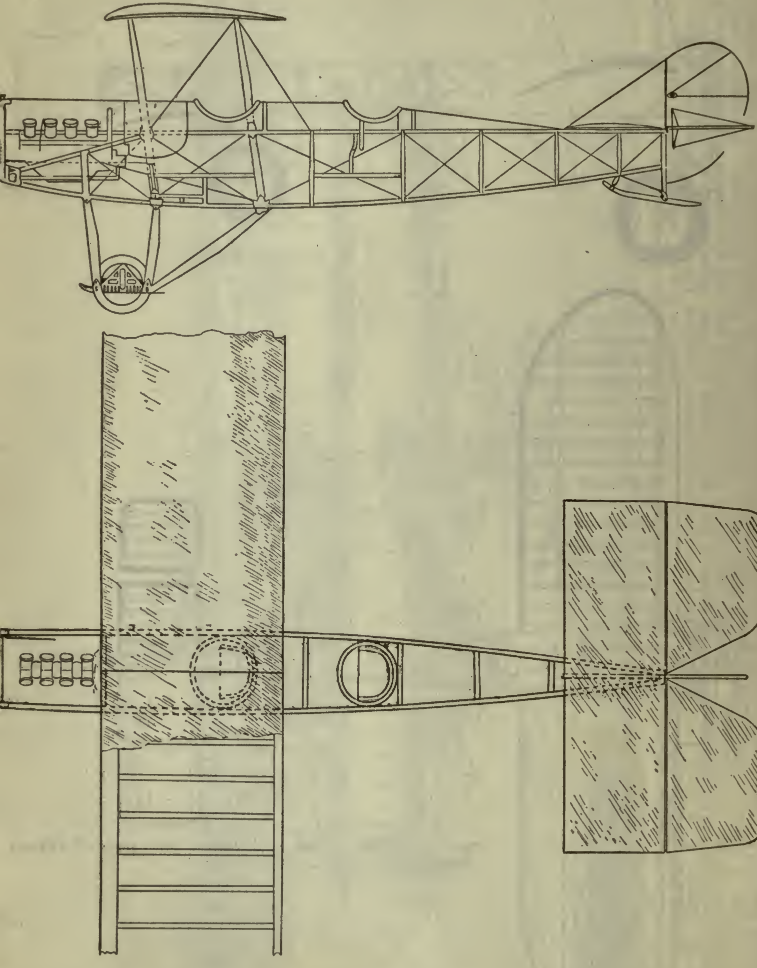


FIG. 2,946.—Curtiss hydro-aeroplane. This model was designed to meet the Navy's requirements for a water training machine. In its general lines it follows closely the land type of military tractor, except that it is, of course, fitted with pontoons in place of wheels. It is fitted with a single Curtiss OXX engine which develops 100 horse power at 1400 *r.p m.* Its maximum wing span is slightly in excess of 53 feet, and its overall length is 29 feet 10 inches. It weighs empty 1,900 lbs. and will carry 510 lbs. of useful load. In recent tests held at Newport News it attained a maximum horizontal speed of 70 miles per hour, a minimum flying speed of 45 miles per hour, and climbed 2,000 feet in 10 minutes. A hydro-aeroplane is peculiarly adapted to scouting purposes and due to its compactness may readily be "knocked down" and stored on ship. It resembles the land machine in general form and in the arrangement of the control.



FIGS. 2,947 and 2,948.—Elevation and plan of Bleriot monoplane.



Figs. 2,949 and 2,950.—Plan and elevation of a biplane frame showing construction. In fig. 2,949 the lower wing is not shown for clearness.

Here is seen the lower and upper longerons which may be compared to keelsons and clamps in boat construction. Connecting the upper and lower longerons are upright struts, each panel thus formed being braced by diagonal or cross wires.

At the rear end or tail is a post, exactly similar to a stern post in boat construction, except that it projects downward below the frame and has pivoted at its end the *tail skid*, which cushions the shock in landing.

The longerons terminate in front at the radiator, the construction being very strong as the engine is placed just back of the radiator as in an automobile.

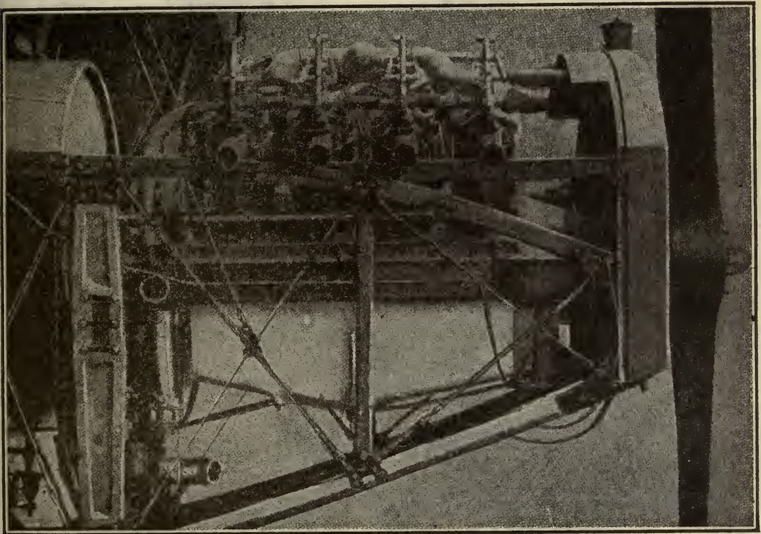


FIG. 2,951.—Front end of frame showing bracing at ends of longerons, engine bed and fastening for radiator.

NOTE.—Nature of wood under stress. Wood, for its weight, takes the stress of compression far better than any other stress. For instance, a walking stick of less than 1 pound in weight, if held perfectly straight, will probably stand a compression stress of over 1,000 pounds before crushing, whereas if the same stick be put under a bending stress, it will probably collapse under a stress of, say 50 pounds. Hence, in aeroplane construction the design is such as to keep as far as possible the wooden compression members in direct compression.

On each side of the frame, just back of the engine space, and fastened to the upper and lower longerons are fore and aft struts to which the wings are attached. These struts are made rigid by cross, and fore and aft diagonal brace wires. These four struts are attached at the top to the center wing section.

Attached to each lower longeron are three struts suitably braced forming the landing gear or under carriage which carries the wheels upon which the machine rests when landing, or not in flight.

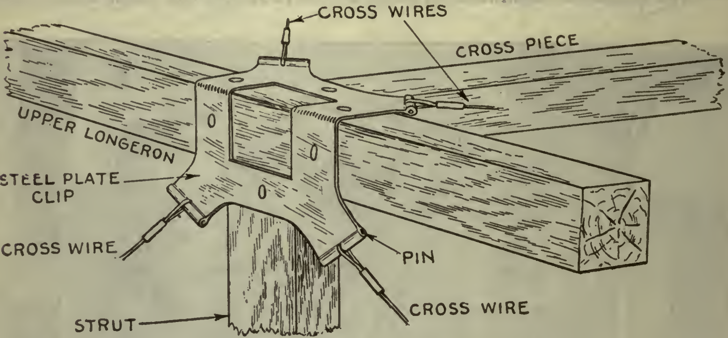


FIG. 2,952.—Detail of upper longeron showing fitting connecting longeron, strut, cross piece and brace wires.

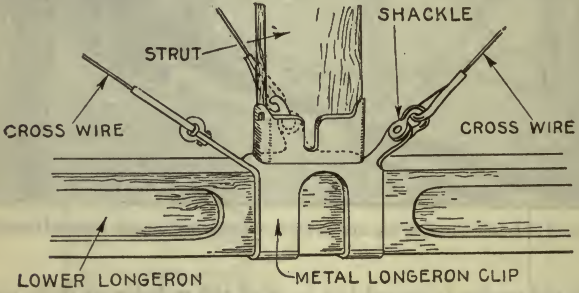


FIG. 2,953.—Detail of lower longeron and strut end, showing fitting for securing strut to longeron and eyes for attaching brace wires.

The illustrations show a two seater, the cockpit openings being seen above the upper longerons and the floor beams between the upper and lower longerons.

Various details in the construction of the frame are shown in the accompanying figures.

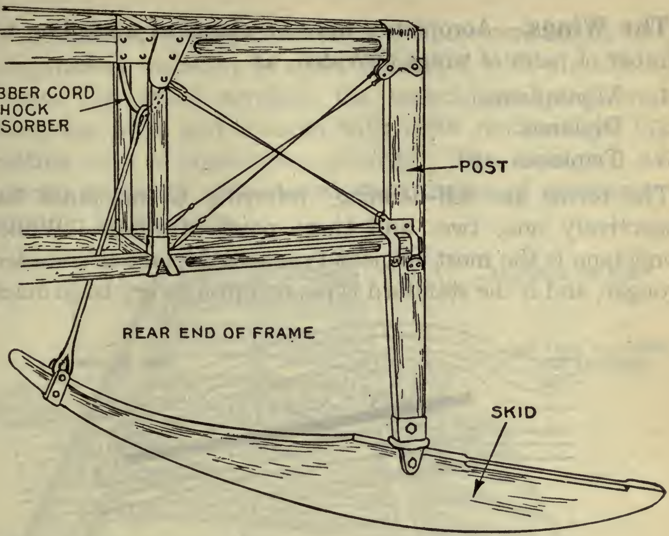


FIG. 2,954.—Rear end or tail of frame showing rear post and tail skid.

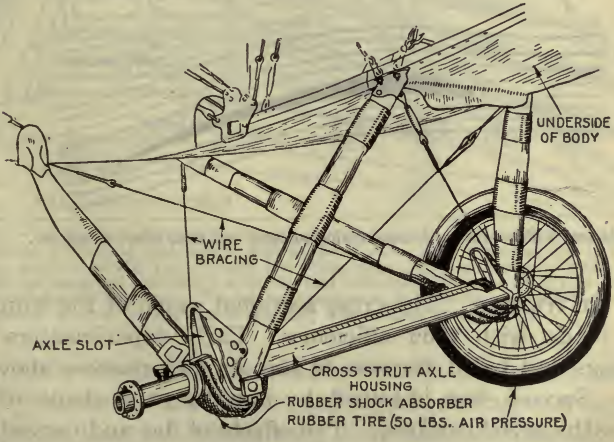
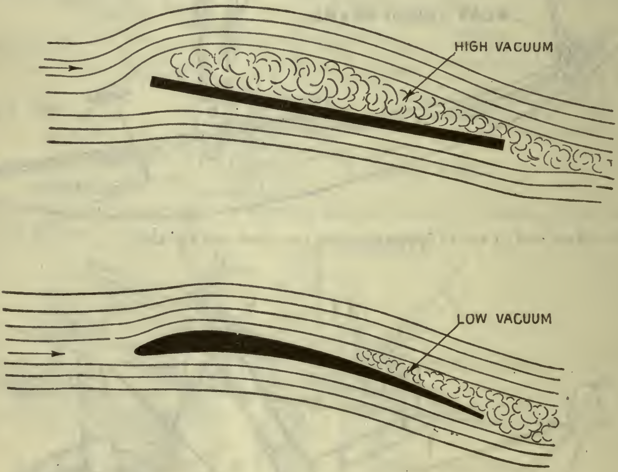


FIG. 2,955.—Landing gear or under carriage, showing two strut type, attachment to lower longerons, wire bracing, shock absorbers, etc.

The Wings.—Aeroplanes may be classified according to the number of pairs of wings provided, as

1. Monoplanes.
2. Biplanes.
3. Triplanes, etc.

The terms are self-defining, referring to machines having respectively one, two, and three pairs of wings. While the monoplane is the most efficient type, the biplane is considerably stronger, and is the standard type, except for very large machines which are triplane. The cross sectional shape of the wings has much to do with their efficiency. Early experimenters tried at wings and failed because of the air disturbances above the wings. Success was obtained by imitating the shape of bird wings with curved surfaces. The effects of flat and curved wing sections are shown in figs. 2.956 and 2,957.



IGS. 2,956 and 2,957.—**Flat and curved wing surfaces and their effect on the air.**

In construction a wing consists of a framework made up of two longitudinal members or spars connected with a number of light cross ribs called aerofoils, the assemblage being braced diagonally by wires and covered with cloth rendered air tight by treating with a preparation or *dope*. The dope acts as a filler giving a smooth surface and some degree of resistance to deterioration by exposure.

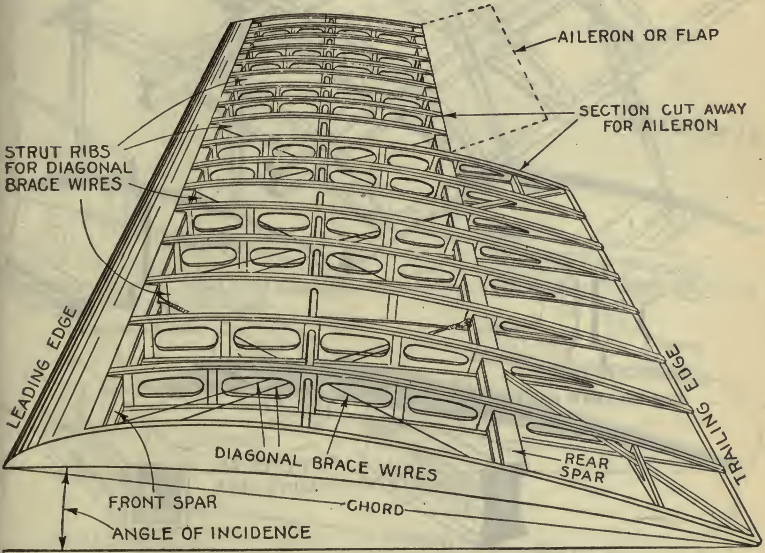


FIG. 2,958.—Construction of wiring framework showing the front and rear spars, ribs, diagonal wire bracing, location of control flap, with names of parts, etc. On either side of the spars run thin strips of spruce, bent to form the proper wing curve on top and bottom. The main ribs are separated and held in shape by thin webbing between the outer ribs, this being cut full of holes for lightness. The leader edge may be a U-shaped spar of wood, metal or veneer, while the rear trailing edge is generally wire or small steel tubing.

Fig. 2,958 shows the construction of the framework just described with names of the parts and terms relating to wings. It will be noted that part of the wing is cut away at the end for the movable control flap or *aileron*, which is hinged to the rear spar.

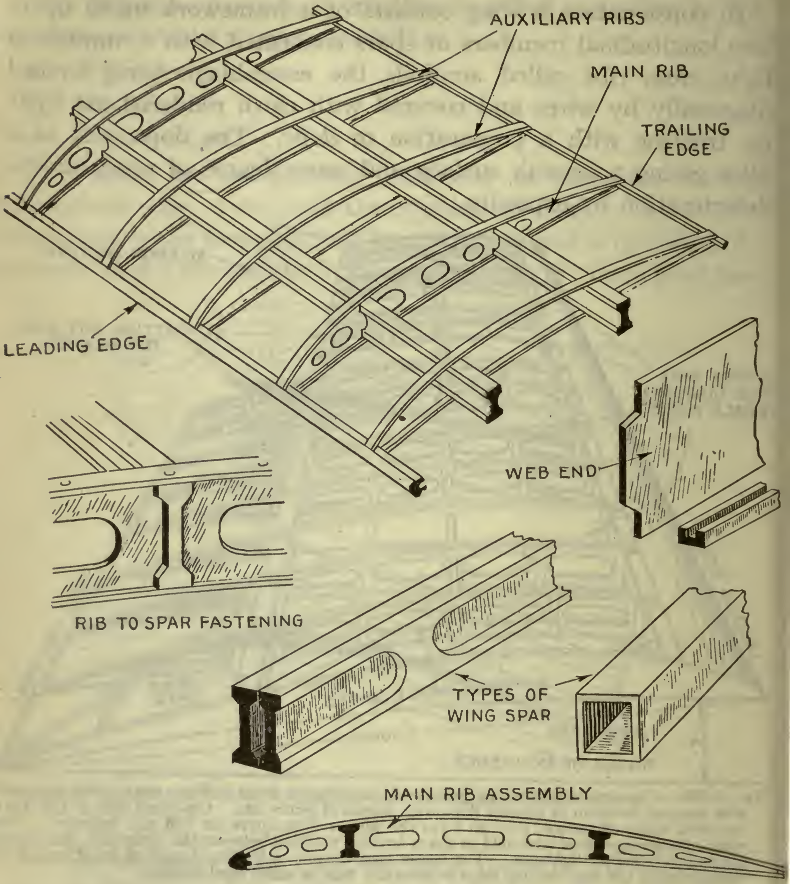
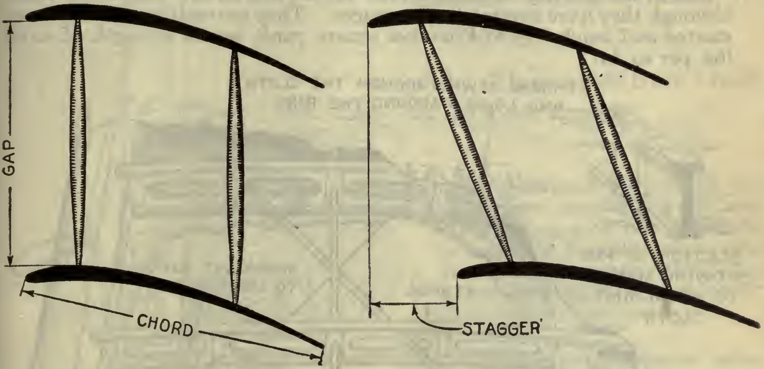


FIG. 2,959.—Wing construction with main and auxiliary ribs. ***In this arrangement*** there are a certain number of ribs between the main ribs which have no support other than the spars or narrow chord sections, and by spaces or extra wing spars on wide chord wings.

FIGS. 2,960 to 2,966.—Various details of wing framework construction. Fig. 2,960 rib to spar fastening; figs. 2,961 and 2962, web end; fig. 2,963, rib section; figs. 2,964 and 2,965, types of wing spar; fig. 2,966, main rib assembly

Figs. 2,960 to 2,966 show details of the framework construction.

On multi-plane machines where one wing is superimposed upon another, the wings are connected together by struts attached to the wing spars as shown in figs. 2,967 and 2,968, and fixed by diagonal wire bracing. In this arrangement it will be noted that the *chords* are parallel, and that the upper wing is in advance of the lower wing, this difference in position projected being called the *stagger*, also the distance between the planes is known as the *gap*.



FIGS. 2967 and 2968.—Method of connecting wings by struts and diagonal bracing, illustrating the terms *chord, gap and stagger.*

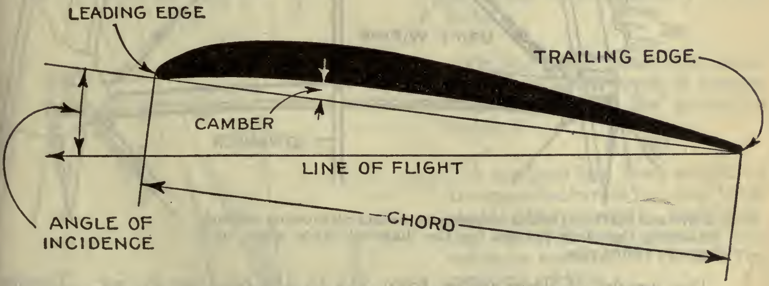
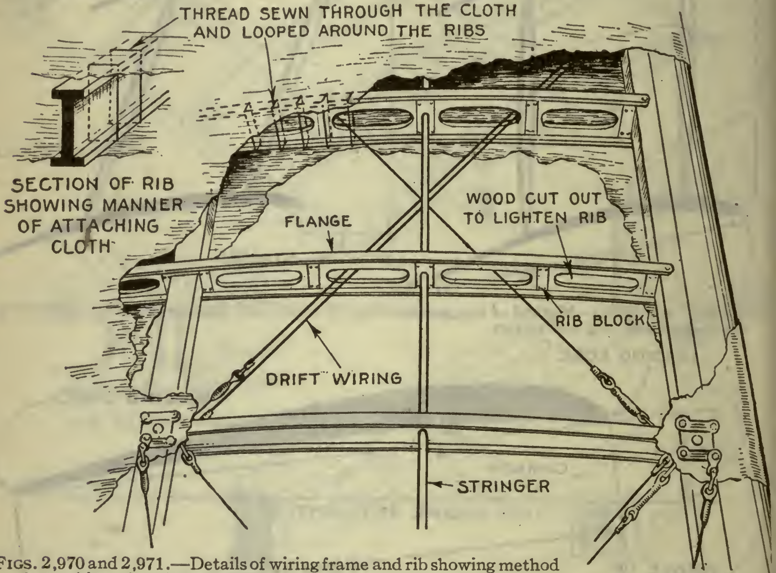


FIG. 2969.—Wiring main rib illustrating the terms *camber, chord, angle of incidence.*

In practice, the best results are obtained by making the gap equal to the chord; if made greater the added length and head resistance of the struts and wires, between the planes outweigh the advantages of increased efficiency.

Wing Covering.—Of the different fabrics used for covering the wings, linen is the most suitable because it is strong, tough, and does not easily tear.

Due to the scarcity of linen, woven cotton fabrics are extensively used, although they have several disadvantages. They are easily torn when once started and weigh 1½ to 4 ozs. per square yard; tensile strength 25 to 80 lbs. per sq. in.


THREAD SEWN THROUGH THE CLOTH AND LOOPED AROUND THE RIBS
SECTION OF RIB SHOWING MANNER OF ATTACHING CLOTH
FLANGE
WOOD CUT OUT TO LIGHTEN RIB
RIB BLOCK
DRIFT WIRING
STRINGER

FIGS. 2,970 and 2,971.—Details of wiring frame and rib showing method of attaching the cloth fittings for the diagonal brace wires, and stringer to stiffen ribs.

The weight of linen varies from 3½ to 4¾ ozs. per sq. yd. Tensile strength 60 to 100 lbs. per sq. in., depending on the weight of the cloth.

Dopes.—This vulgarism is a general term applied to the various impregnating compounds used in treating the cloth surfaces to render them air and moisture tight, taut and smooth. It is claimed that the application of dopes increases the strength of the fabric.

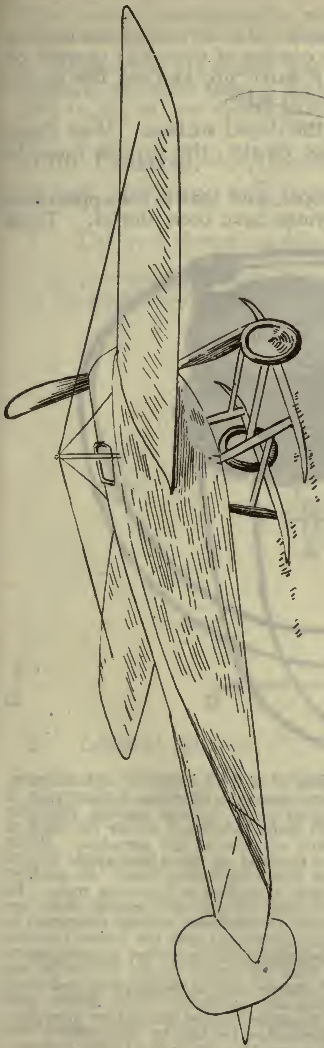


FIG. 2,972.—Hanriot monoplane.

Dopes are divided into two classes.

1. Those made from a base of cellulose nitrate or pyroxylin.

2. Those made from a base of cellulose acetate.

In either case the base is dissolved in a suitable solvent, and often substances are added to preserve the flexibility of the substance and to modify shrinking. These substances are analogous to camphor in celluloid.

The difference between cellulose acetate and cellulose nitrate being the inflammability of the latter.

On the other hand, the initial cost of cellulose dopes is greater than the cost of the pyroxylin dopes.

A medium has been struck by the manufacturers in using three coats of the pyroxylin, and finishing off with two coats of the cellulose acetate. This is a fire resisting surface.

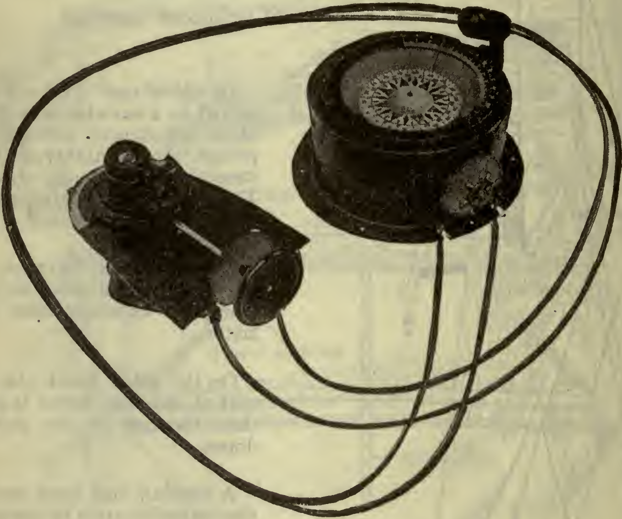
Here is an example of the difference between the two dopes.

Test panels were made up of the fabric stretched on a frame, and the dopes above mentioned applied. After they had dried, this experiment was undertaken.

A lighted match was placed on these panels. In the case of the acetate dope, the fabric was not ignited until the coating of dope had charred or melted. The pyroxylin dope immediately flared up, and set fire to the fabric. Therefore, the cellulose dope is much safer.

Spar varnish is sometimes coated over the doped surface. Most dopes do not stick well on varnish, and to make repairs, the varnish must be removed.

There is one trouble encountered with dopes, and that is the appearance of white spots on the surfaces after the wings have been doped. These



IG. 2,973—Sperry synchronized drift set. It consists of a drift indicator and compass. In the drift indicator a specially powered monocular telescope, affording a clear view of the ground below, is mounted so that it can be turned about on axis passing vertically through it. The telescope contains several parallel hairs, stretched across the field of vision, which normally rest parallel with the fore and aft line of the aeroplane. While in flight, the earth's surface flows so rapidly across the field of vision that single objects or irregularities cannot be identified, but seem to form into long flowing lines. An analogy of this may be obtained by looking at the ground from a window of a moving train. ***To determine the angle of drift,*** the telescope is turned so that the parallel hairs are coincident with the lines of motion. A pointer which travels over a scale indicates, in degrees the actual angle of drift. The compass is so constructed that the lubber line can be moved relatively to the fore and aft line of the aeroplane. This renders it unnecessary for the pilot to make a mathematical deduction of what his course should be, as he has only to move the lubber line through the number of degrees and in the direction indicated on the drift indicator. The indicator is mechanically corrected to the lubber lines shifting mechanism so that the movement of the telescope moves the lubber line thus correcting any drift. Thus the pilot has his course prescribed for him by the observer who operates the drift indicator, which at the same time moves the lubber line in the compass.

spots occur most frequently with the pyroxylin dopes, rather than with the acetate dopes. These spots show up.

Dopes cover 2½ to 5 sq. yds. per gal. when applied so as to give a durable surface, generally four to six coats. The weight of the dope ranges from 1½ to 3½ oz. per square yard.

The Control.—An aeroplane in flight is subject to three motions, the same as a vessel in a rough sea.

1. Up and down motion;

In pointing upward or downward, as in climbing or descending, similar to the angles of inclination taken by a vessel in riding a wave.

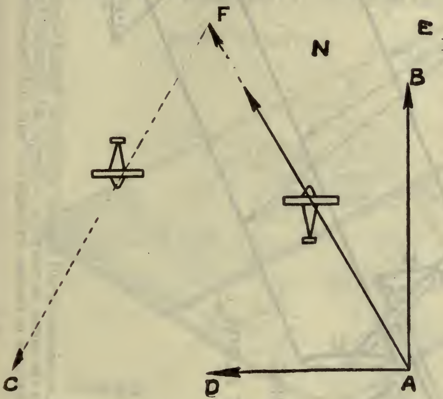


FIG. 2,974—Effect of drift. Point A, represents the point of departure. E, is the point to be reached, and AD, is a side wind whose velocity compares with the velocity of the aeroplane as AD, is to AB. It will be seen from this that although the pilot's compass is indicating north, his true course is considerably west of north, along the line AF, and he will miss his objective, E, by many miles. If, upon failing to find his objective, the pilot turn about to return to his base, it is natural for him to steer south or in the direction from which he came. This would carry him along a line parallel to CF. In cases where it is important to arrive at or return to a given place, it is not difficult to appreciate the value of such an instrument, for determining deviation to offset drift.

2. Lateral motion;

Due to changing air conditions one wing may tip upward and the other downward tending to upset or turn over the machine, just as a vessel rolls from one side to the other in a rough sea.

3. Directional motion;

The machine may change its course either to right or left, as a vessel responds to the rudder.

Because of these possible motions and in order that the operator may control the motion of the machine by causing it to assume any desired position it is necessary that there be

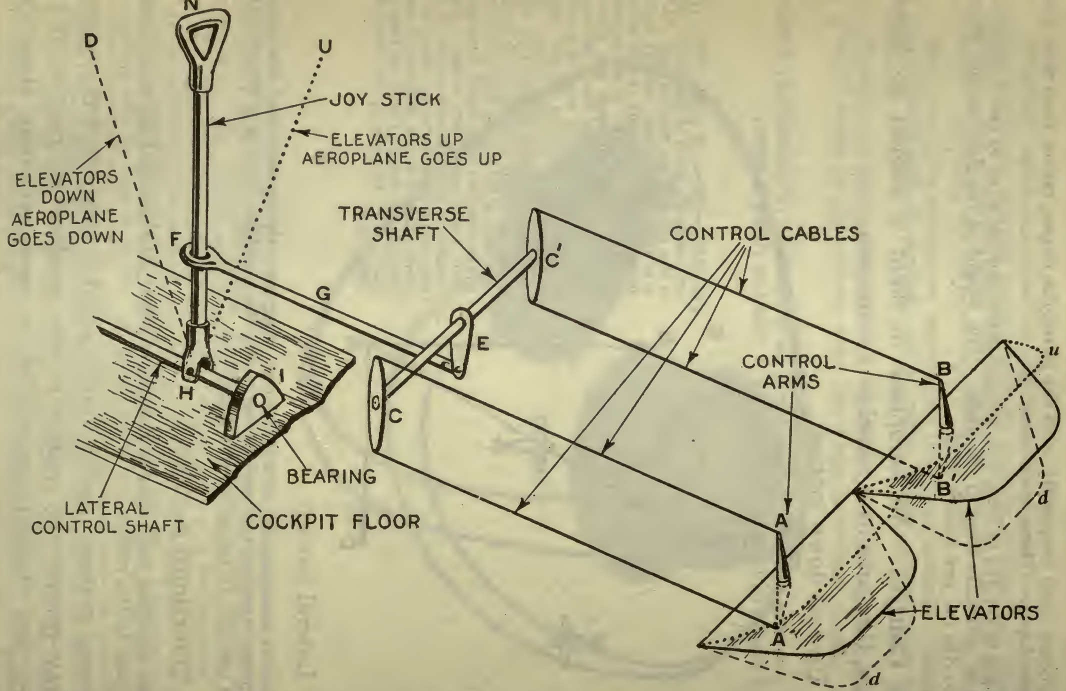


FIG. 2,975.—The up and down control, consisting of tail flaps or "elevators," hinged across the tail post and connected by suitable gears to the joy stick so that as the latter is moved in a fore or aft direction the elevators are respectively inclined downward or upward, causing the machine to change its angle of flight and respectively point downward (to descend) or upward (to climb).

provided controlling devices for each motion. These are respectively:

1. The up and down control;
2. The lateral or rolling control;
3. The right and left control;

These three controls are the *movable* controls as *distinguished* from the *fixed* controls or *stabilizers*.

The Up and Down Control.—At the upper end of the tail post where the upper longerons are attached is a transverse piece placed at right angles to the tail post, and to which the up and down control flaps or elevators are hinged as shown in fig. 2,975, one on each side of the post.

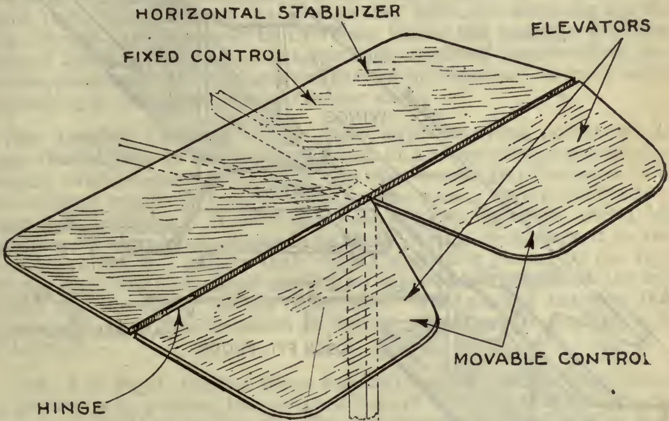


FIG. 2,976.—Horizontal stabilizer or fixed horizontal control surface to prevent sensitive response to the elevators. If the stabilizer be too large the machine becomes stiff and unduly resists the elevator control; if too small, it becomes tricky. ***In practice,*** it has been found that a stabilizer having an area of 6 to 8% of the wing area and placed from 2 to 2½ chord lengths back of the wing gives the best results.

Each elevator has two control arms AA′ and BB′, connected by cables to the arms C and C′, the latter being attached to a transverse shaft. Keyed to the center of this shaft is an arm E, which is connected to the joy stick at F, by the rod G.

The joy stick is pivoted at H, to the lateral control shaft, which is free to turn in the bearing I, giving virtually a universal joint at H.

In flight when the joy stick is in the vertical or neutral position N, the

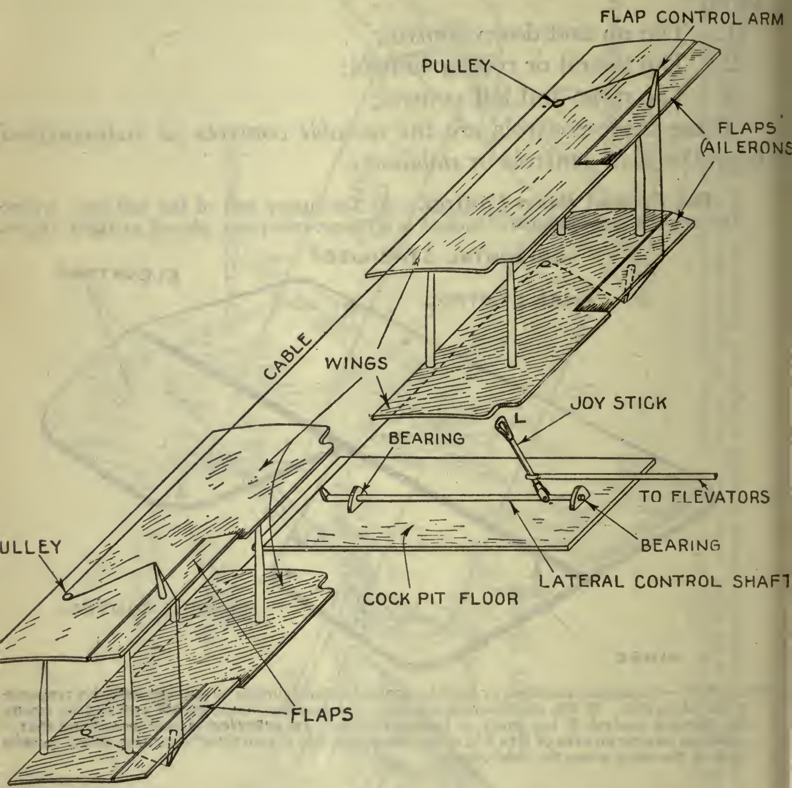


FIG. 2,977.—The lateral control. It consists of movable flaps, hinged to the trailing side of the wings at the ends as shown, and connected to the joy stick by suitable cable and pulley gearing. The gear is so arranged that the flaps on one side move in an opposite direction to those on the other side. On some machines only the top wing is provided with flaps.

elevators are in a plane perpendicular to the tail post and do not tend to change the normal flying angle of the machine.

Moving the joy stick fore or aft moves the elevators respectively up or down. Thus when the stick is moved forward to position D, the elevators are inclined downward to position *d*, giving an upward pressure on the elevators, steering the machine *down*. Again moving the stick aft to position U, deflects the elevators upward to position *u*, giving a downward pressure on the elevators and causing the machine to point upward.

Evidently if some means were not provided to resist somewhat the effect of the elevator control the machine would be so sensitive to the movement of the elevators as to become unstable or tricky, hence a fixed tail surface called the horizontal stabilizer is provided as shown in fig. 2,976. The effect of this fixed surface is to hold the machine in its normal flying angle, thus giving inherent stability so that when the elevators are moved this inherent stability must be overcome before the machine will alter its angle of flight. The machine is accordingly rendered less sensitive, that is, the stabilizer tends to damp out oscillations or "hunting" of the machine in responding to the movement of the joy stick.

The Lateral Control.—If in flight any disturbing condition should cause the machine to lose its equilibrium in a transverse direction, that is, unequal wind pressures may cause one wing to rise and the other to drop, the machine would roll over or upset, unless means were provided to overcome such tendency. Accordingly, each wing is provided with a hinged flap near its elevator as shown in fig. 2,977, being so connected to the joy stick by cables and pulleys that when the stick is moved to the right or left the flaps will be inclined *upward* on the side toward which the stick is moved and *downward* on the other side. The gearing between the joy stick and flaps is clearly shown in the figure so that its working is easily understood.

Now, if in flight, the left wings rise and the right wings drop, this may be overcome by pushing the joy stick toward the high side (that is, to the position marked L in the figure). This elevates the left flaps causing pressure on that side from above, and depresses the right flaps causing pressure there from below. The left side then is pushed down and the right side up, bringing the machine back in a horizontal position.

When the horizontal position is reached the joy stick is brought back to

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NOTE.—***Warp control.***—In this method of lateral control, wires pass from the control stick or wheel through fan heads, low down on the body, to the rear outer strut of the warping section of the way. There are various ways of attaching these warping wires. One method is to pass the wires through fan leads fitted at the bottom of the rear outer struts to different points along the warping section of the top plane. Only the top plane is warped, and may be warped along the entire length of the plane or only the outer section. Warp controlled machines are fitted with a compensating wire which is usually placed along the top of the upper plane leading from one warp section to the warp section on the opposite plane. By this means when one warp section is pulled down by the control device, the opposite warp section is pulled up by the compensating wire.

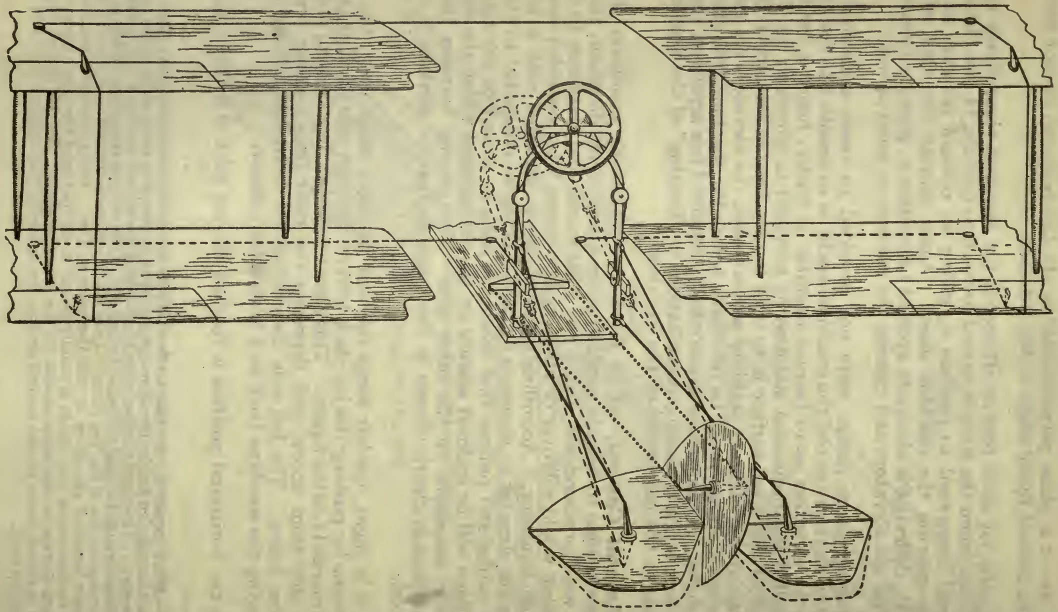


FIG. 2,978.—Wheel or "Dep" (Deperdussin) control. It works like the steering wheel of an automobile to control the ailerons or flaps, and is tilted fore or aft to operate the elevators. This type of control is adopted to slow acting and especially large machines where a geared down leverage for the wing flaps is necessary. On some machines the wheel is attached to a central post instead of the arch.

the vertical or neutral position. With the connection as shown the joy stick is naturally moved toward the side to be lowered.

Lateral Stability.—This may be defined as *the sideways balance of the machine*. There are several conditions which tend to turn a machine over sideways, or cause it to "skid" These are:

1. Excess wind pressure on one side.
2. Sudden gust of wind not in the direction of the line of travel.

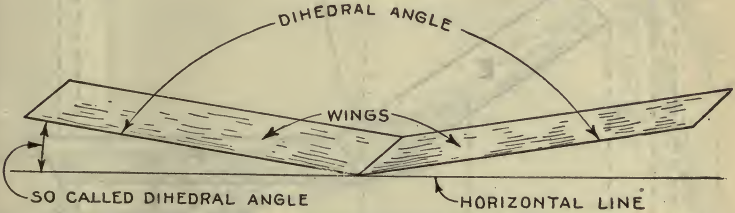


FIG. 2,979.—Inclination of the wings to the horizontal to secure lateral stability. The dihedral and so called dihedral.

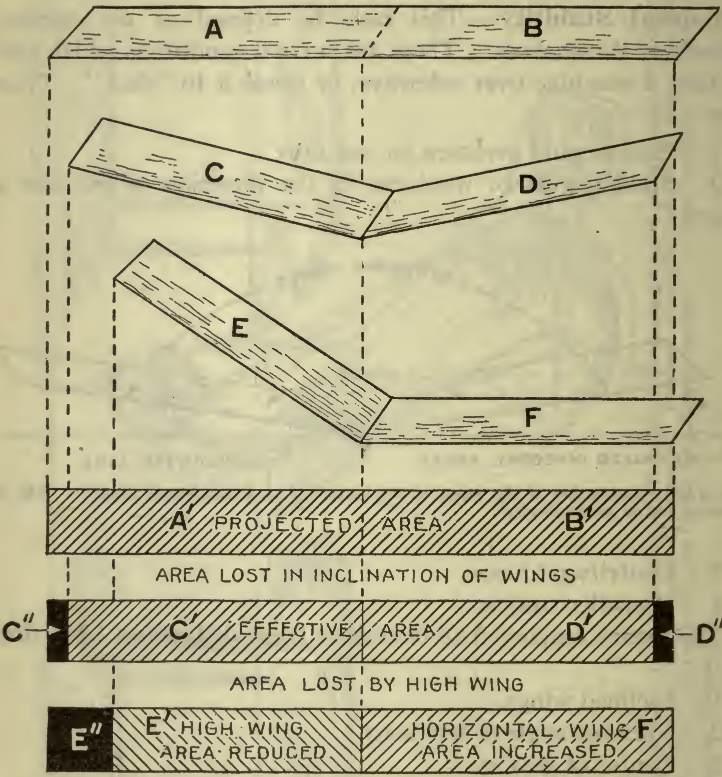
3. Centrifugal force.
4. Propeller torque.

The means employed to overcome these disturbing conditions are:

1. Inclined wings.
2. Variable angle of incidence.
3. Center of gravity.
4. Side surface (so called keel surface).
5. Vertical stabilizers.

Inclined Wings.—To secure lateral stability, the wings are inclined upward from the frame to their tips as shown in fig. 2,979, the angle between the two wings being known as the dihedral and not the angle which the wings make with the horizontal as conveniently and erroneously stated by some writers.

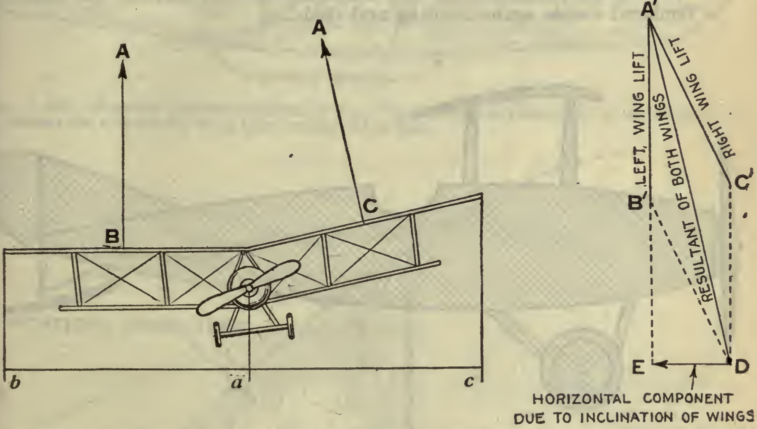
The effect of inclining the wings in giving lateral stability is shown in figs. 2,980 and 2,985.


A
B
C
D
E
F
A′ PROJECTED AREA B′
AREA LOST IN INCLINATION OF WINGS
C″
C′ EFFECTIVE AREA D′
D″
AREA LOST BY HIGH WING
E″
E′ HIGH WING AREA REDUCED
HORIZONTAL WING AREA INCREASED F

IGS. 2,980 to 2,985.—Inclination of wings; lifting and stabilizing effects. Since the lifting power of wings depends on their *projected* area instead of their *actual* area, it follows that when the two wings A and B, are in the same plane, their projected area A′,B′, will equal their actual area A,B, and the full lifting power will be obtained. Now in the case of inclined wings C,D, the projected areas C′,D′, are less than their actual areas C,D, and their lifting power is reduced. Here, C″,D″, represent the projected areas lost by the inclination, which also represent the loss in lifting power. ***Stabilizing effect:*** If in flight a sudden gust of wind cause the machine to tip to one side, so that the wings assume the positions E,F, then the projected area E′, is less than F′, the projected area of F, hence the lifting power of F, is greater than that of E, which tends to bring the machine back to its horizontal position. The solid black area E″, is the projected area lost by wing E, when rotated from the horizontal position A, to the inclined position E, area F, being equal to D′+D″.

Now when the wings are inclined, the effective lifting surface is equal to the projected area, and as shown in fig. 2,980, when the machine is not inclined to either side, the projected area of each wing is the same, hence the lifting force on each wing is the same.

Now, if the air pressure become greater on one side than the other, as by a sudden gust, it will raise that side as shown in fig. 2,981. When the machine becomes inclined sidewise as here shown the projected area of the low wing is increased, and that of the high wing decreased, consequently the lifting force on the high side is reduced and that on the low side increased.



FIGS. 2,986 and 2,987.—Inclination of wings causes a side thrust. Let **AB**, and **AC** (=*ab* and *ac*) equal lift of the two wings, then taking **A′B**, and **AC′** equal and parallel respectively to **AB** and **AC**, completing the parallelogram, and draw **A′D**, the resultant. Since this upward pull **A′D**, is inclined to the vertical, there will be a side thrust equal to the horizontal component **DE**, acting in the direction **DE**, to push the aeroplane sideways.

These unequal forces will clearly cause the low wing to rise and the high wing to descend, thus bringing the machine back to a horizontal position. The wings are inclined (to the horizontal) only a few degrees, otherwise their lifting power would be unduly reduced.

Center of Gravity—The position of the center of gravity has a marked effect on the lateral stability of the machine.

In order to produce a lateral righting effect the center of gravity should be low, just as the ballast in a sail boat is placed low to counteract the side thrust of the sails. However, if the center of gravity be too low, the side oscillations or swings are liable to be increased in amplitude; if too high, the tendency would be to upset the machine.

Side Surface.—It is difficult to understand how the idiotic expression *keel surface* ever came into use to mean the *side surface* of an aeroplane.

Comparing an aeroplane to a boat, any one knows that the keel and side surface of the boat are two entirely distinct things, the surface of the keel being extremely small as compared to the surface of the side.

The term *side surface* means everything to be seen when viewing an aeroplane from the side, that is, by definition *the projected area of everything visible from the side of an aeroplane.*

By properly proportioning and arranging the side surface, the machine is rendered stable against rolling and skidding

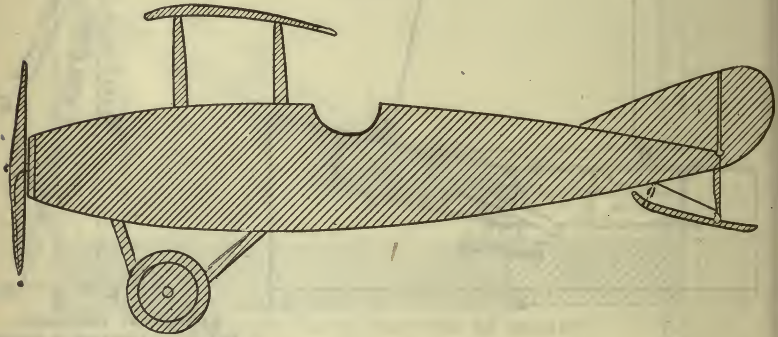


FIG. 2,988.—The side surface of an aeroplane, being *the projected area of everything visible as viewed from the side*, it is the ***effective surface upon which the wind may act from the side.*** The term *keel surface* is frequently and ill advisedly used for side surface.

If the side surface be low, the side force, as of a side gust of wind blowing against it, will rotate the aeroplane sideways so that the windward wing will sink; if high, the windward wing will rise, but if on the same level with the center of gravity, there will be no tendency to rotate the machine, and will simply oppose the skidding.

Vertical Stabilizers.—These are the small fins placed above the center of gravity 1, on the tail, and 2, sometimes on top of the upper wings. If, due to a sudden side gust, the machine should move sideways to any extent, the high fins resist this movement and also, because they are higher than the center of gravity, tend to *bank up*, the machine, that is, revolve it on its longitudinal axis so that the windward wing sinks, thus giving lateral stability.

Evidently, if the vertical stabilizers were placed below the center of gravity the opposite rotary effect would be produced which would tend to upset the machine.

The rear fin placed as an advance continuation of the rudder, evidently prevents over sensitiveness to the movement of the rudder thus avoiding hunting or directional oscillations of the machine in responding to the rudder.

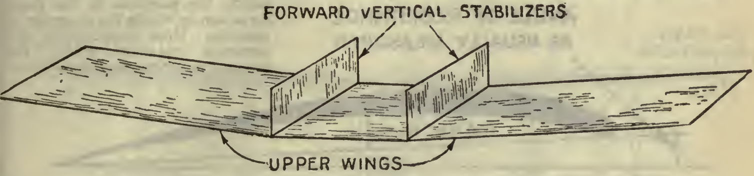


FIG. 2,989.—Forward vertical stabilizer or fin. These are used principally in hydroplanes to balance the abnormally large side surface of the body.

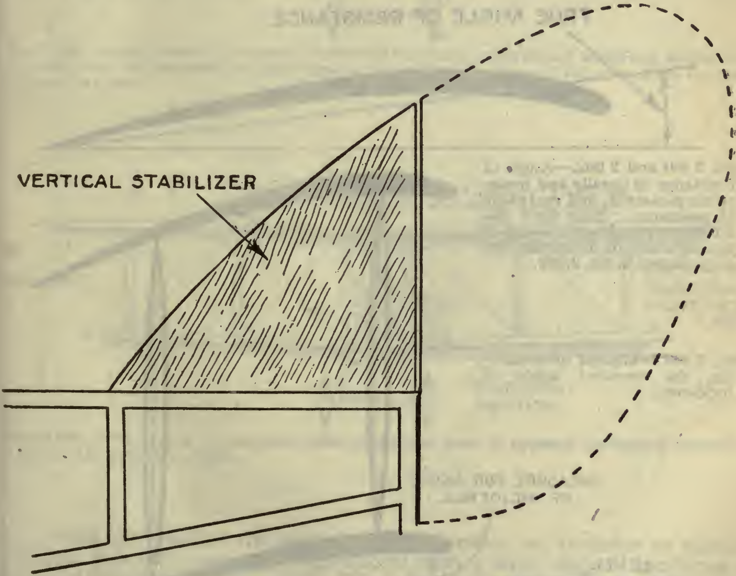
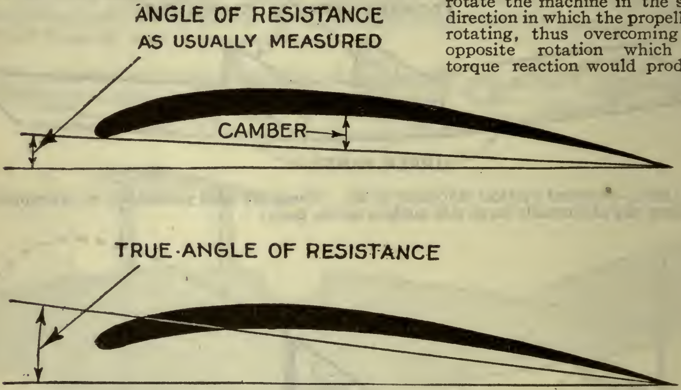


FIG. 2,990.—Rear vertical stabilizer or fin, to secure stability against skidding by resisting the skid, giving a banking effect, also to damp out the directional oscillations due to response to rudder, and to overcome centrifugal force, or tendency of the tail to skid outward in making a turn.

Variable Angle of Incidence.—If the propeller were held so that it couldn't turn, the power of the engine would be spent in turning the aeroplane round its shaft as an axis.

In fact, the air striking the blade of the propeller obliquely presents considerable resistance against its rotary motion, and a corresponding force is exerted on the machine to turn it in an opposite direction. To overcome this tendency to turn the machine, it suffices to give a greater angle of incidence and therefore greater lift to the left wing, assuming the propeller to be turning clockwise from in front. This increased angle of incidence tends to rotate the machine in the same direction in which the propeller is rotating, thus overcoming the opposite rotation which the torque reaction would produce.



FIGS. 2,991 and 2,992.—Angle of resistance as usually and incorrectly measured, and real angle of resistance. This error has crept in no doubt on account of the difficulty of measuring the angle shown in fig. 2,992.

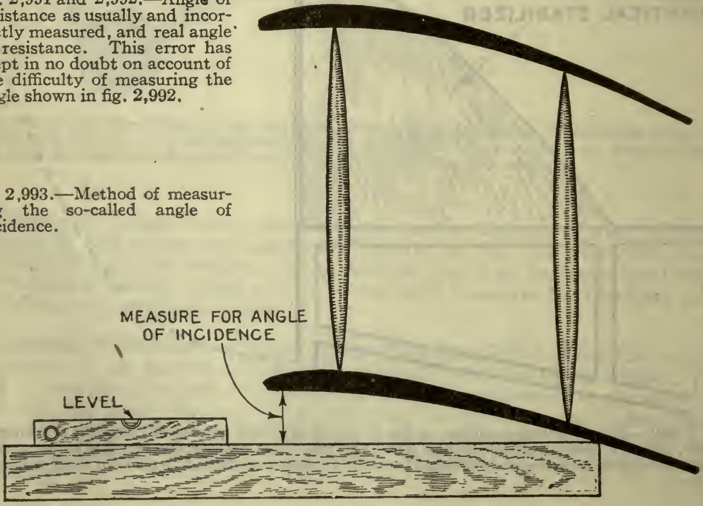


FIG. 2,993.—Method of measuring the so-called angle of incidence.

Two rather objectionable terms are used to indicate the variable angle of incidence provided for overcoming the propeller torque: ***Wash in*** meaning the increase of angle of incidence toward the wing tips, and ***wash out*** meaning the decrease of angle of incidence toward the wing tips, as shown in fig. 2,994. Both lateral and directional stability may be improved by washing out the angle of incidence on both sides.

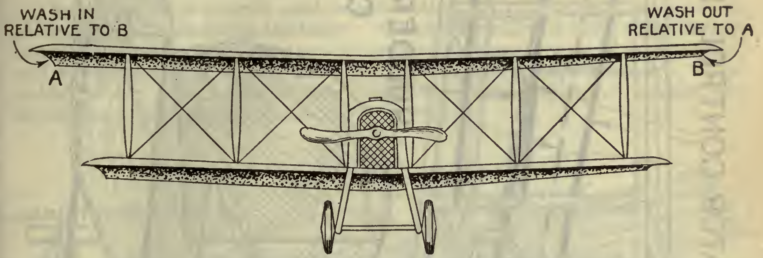


FIG. 2,994.—Front view of aeroplane illustrating the terms ***wash in*** and ***wash out*** which mean respectively *the* ***increase*** *and* ***decrease*** *in the angle of incidence of the wings from the center toward the tips.*

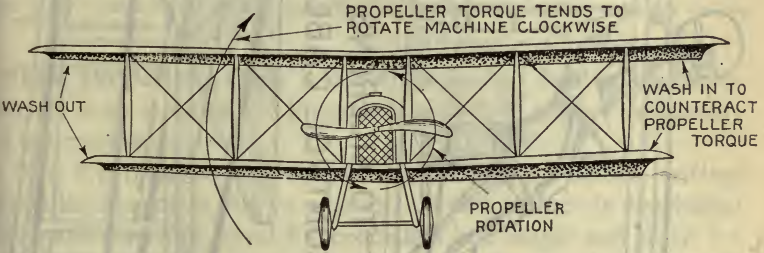


FIG. 2,995.—Front view of aeroplane illustrating how wash in opposes the turning over effect due to the propeller engine.

Wash out renders the ailerons more effective, as, in order to operate them, it is not necessary to give them such a large angle of incidence as would otherwise be required. The effect of ***wash in*** in opposing the tendency of the propeller torque to turn the machine over is shown in fig. 2,995.

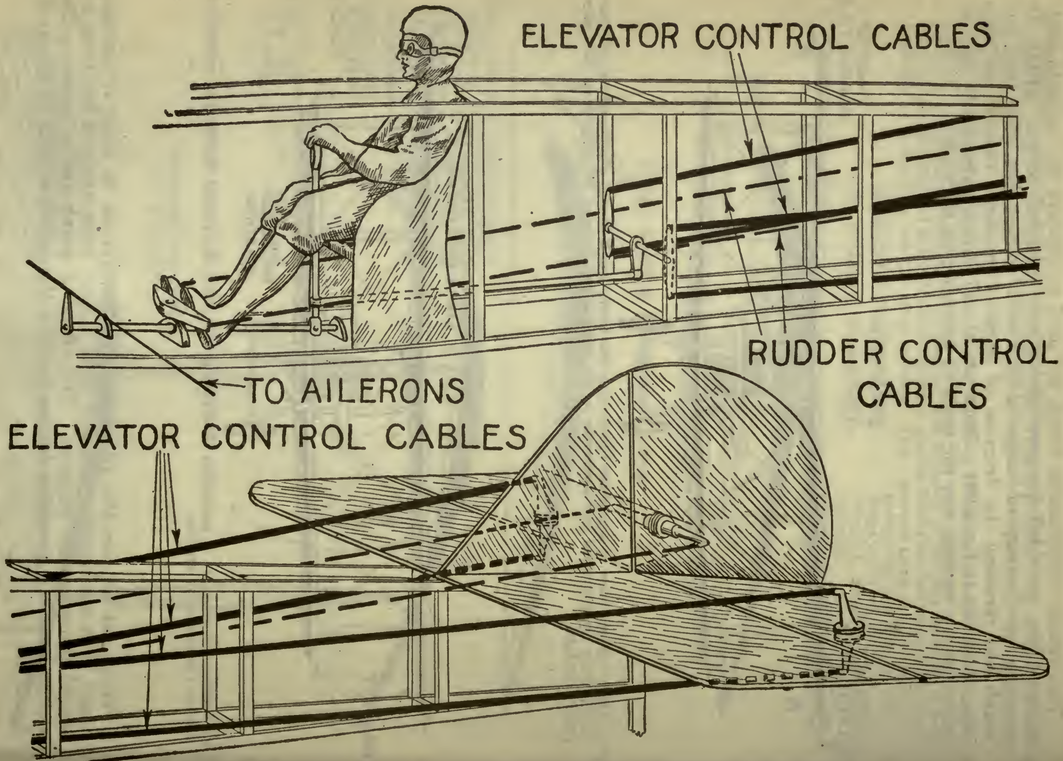


FIG. 2,096.—Rear portion of aeroplane body showing rudder and rudder control as operated by the feet.

The advantage of wash in is offset somewhat by loss of lift, as the lift decreases with the decreased angle.

Directional Control.—An aeroplane is steered by a rudder just like a boat. The rudder is pivoted to the rear post and is connected by cables to a foot bar in front of the operator, so that steering (right or left) is done by means of the feet pressing against the foot bar which is pivoted at the center to the cockpit floor as shown in fig. 2,966.

The rudder is sometimes used in connection with the ailerons as a means of checking any swerving tendency. The ailerons at the same time then in restoring lateral balance, tend to change the directional course of the machine, and the rudder in such case must be used to oppose this tendency. The rudder is also frequently used to preserve a straight course against the disturbing action of side winds.

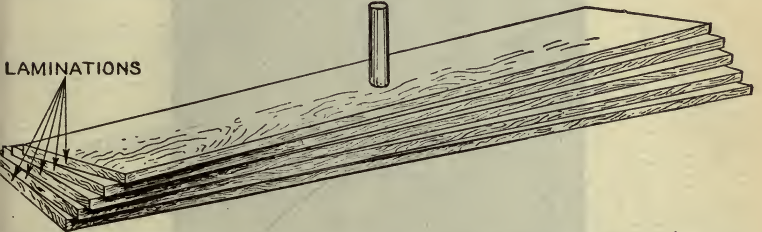


FIG. 2,997.—Assembly of wooden laminations or boards glued together and ready for the routing operation.

The Propeller.—This is hand made, and hence the reason why it is so expensive. Propellers are made almost exclusively of laminations of wood.

Fig. 2,997 shows the laminations assembled before cutting. Each lamination is cut to pattern. Black walnut is considered he most desirable wood for propellers but maple and birch are also used. Black walnut and spruce are often used in alternate ayers. After the laminations are glued together the assembly s roughed out by hand with different forms of draw knife, but n some of the larger shops special routing machines capable of outing several propellers at one time, as shown in fig. 2,998, are used thus saving considerable time and reducing the expense proportionately.

The operation of a propeller is similar to that of a boat, that is the oblique blade surfaces push back a column of air, the resistance of which gives a forward thrust to the propeller shaft.

If the propeller acted in solid matter it would advance forward each revolution a distance depending on the obliquity of the blades. This is called the *pitch*, but with a substance like air there is some *slip* so that the forward advance per revolution is less than the pitch, the difference between the pitch and the actual advance per revolution is the *slip* usually expressed as a percentage of the pitch. Thus, if a propeller, having a pitch of 10 ft., advance, say, 7½ feet per revolution, the slip is 10—7½=2½ ft. or 25%.

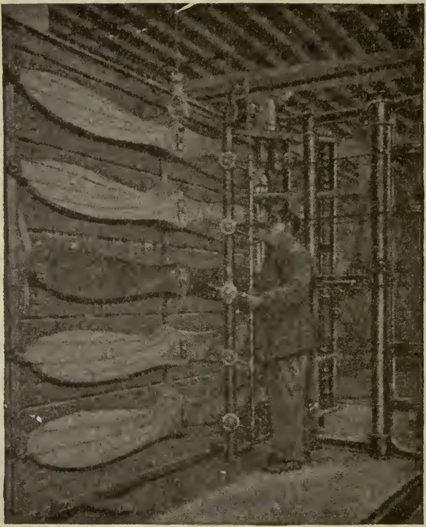


FIG. 2,998.—Multi-tool routing machine for routing several propellers in one operation. The dark propeller in the center acts as a template and guides the four cutting tools through a pantagraph like gear which causes each cutting tool to reproduce the movements of the master tool or guide rod.

In construction, since the outer end of the propeller travels faster than points nearer the center the pitch angle changes, increasing from the end to the hub.

The nearer the ratio of pitch to diameter approaches 1, the more efficient is the blade. The propeller surface should be very smooth especially near the outer end where the speed is greatest, otherwise there will be considerable "skin friction."

The propeller should be perfectly balanced and the surface areas of both ends equal.

Mechanics of Flight

Lift of the Wings.—When the wings are moving through the air, lift is due to: 1, *pressure* against the under surface of the wings caused by the inertia of the air on being deflected downward by the advancing inclined planes, and 2, partial *vacuum* formed over the top of the wing curved surface, also by the inertia of the air by virtue of which it cannot deflect and fill up instantly the space swept through by the advancing inclined wings.

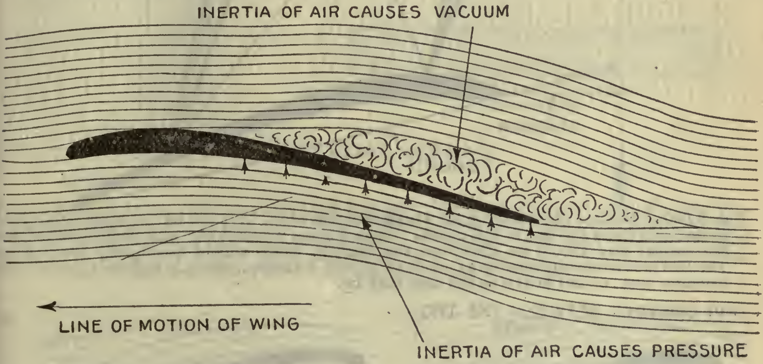


FIG. 2,999.—Section of wing illustrating *lift* as caused by *inertia* of the air in causing upward pressure on the lower surface, and partial vacuum on the upper surface of the wing.

Stagger.—When there are more than one pair of wings as in biplanes, triplanes, etc., if one pair be placed directly over the other pair, the suction region on the top of the under wing, and the compression region below the upper wing both find themselves confined in about the same space, so that the excess

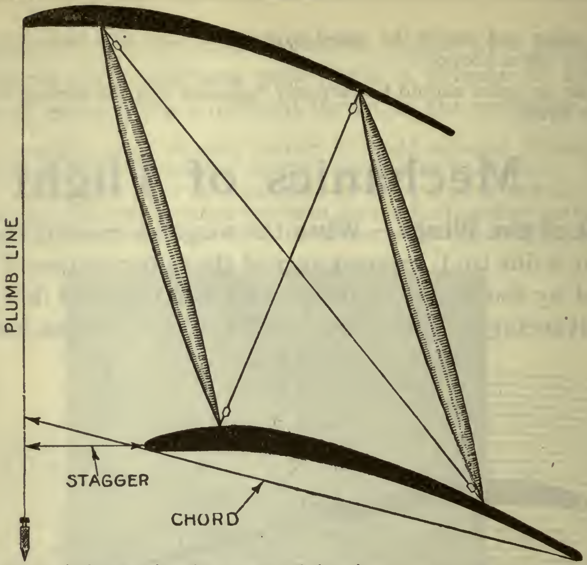
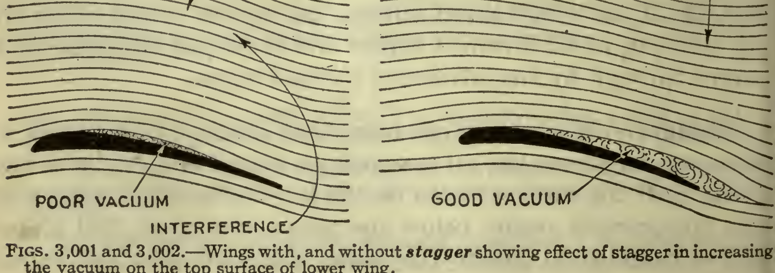


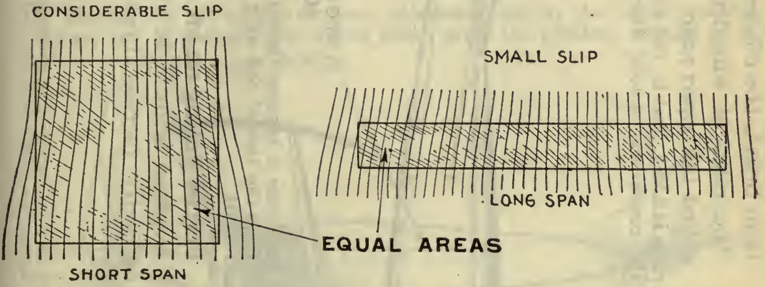
FIG. 3,000.—Method of measuring the ***stagger*** of the wing. This can be measured either horizontally or along the line of the chord. Some makers measure one way, some another, but the correct way should be shown on the diagram which should accompany every wing. The two measurements may be as much as ¼ inch difference, which is sufficient to make the machine nose, or tail heavy as the case may be.



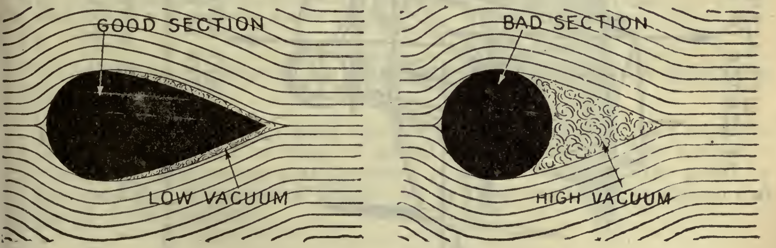
FIGS. 3,001 and 3,002.—Wings with, and without ***stagger*** showing effect of stagger in increasing the vacuum on the top surface of lower wing.

pressure under the top wing causes the air to expand downward and reduce the vacuum on top of the under wing as in fig. 3,001, thus decreasing the efficiency of the wing surface.

It has been found that if the top wing be placed a little in advance of the lower wing (that is, given stagger) there is less interference with the vacuum on top the lower wing, thus the lower wing is rendered more efficient, as shown in fig. 3,002.



FIGS. 3,003 and 3,004.—Wings of short and long span (low and high aspect ratio) showing why the long wing is the more efficient. Wing tips are inefficient because they "spill," or allow the air to slip sideways past their ends, hence in a long span wing as in fig. 3,004 the end portion or tip is only a small percentage of the total area, and accordingly little air slips by the tips as compared with a short span wing which has a wide tip.



FIGS. 3,005 and 3,006.—Good and bad cross sectional shapes of struts. In fig. 3,005, the long easy curve permits the air to follow the surface, thus practically no vacuum is formed; while in fig. 3,006, the air cannot follow the abrupt curve resulting in a vacuum which acts as a drag or resistance opposing its advance.

Wing Shape.—In early experiments it was found that square wings were very inefficient, and if they were made long rather than broad, the efficiency was improved. The ratio of wing length to breadth is called the *aspect ratio*.

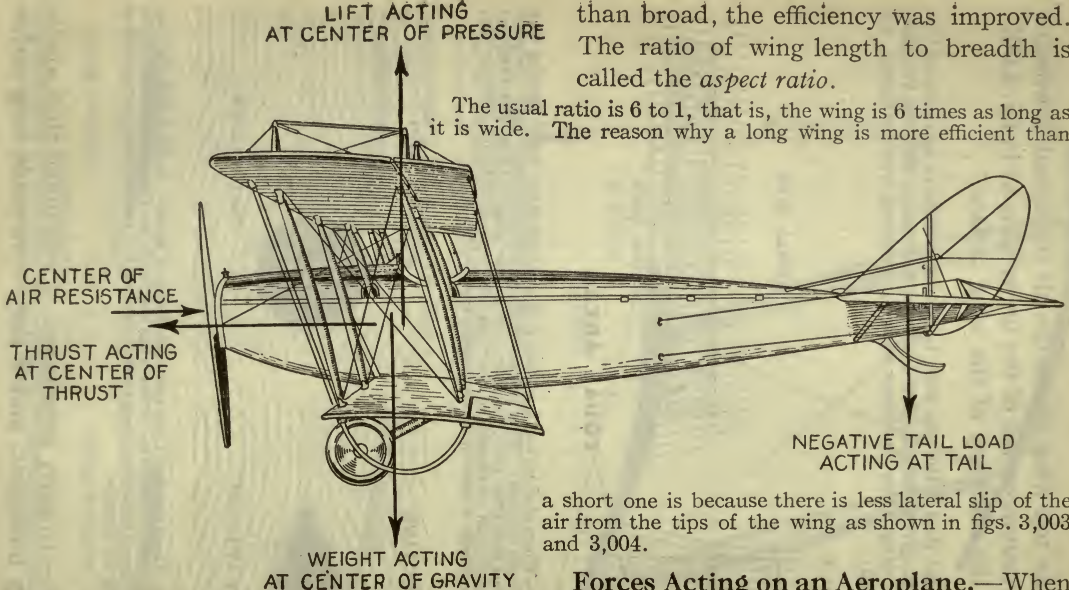


FIG. 3,007.—Equilibrium of forces acting on an aeroplane.

The usual ratio is 6 to 1, that is, the wing is 6 times as long as it is wide. The reason why a long wing is more efficient than a short one is because there is less lateral slip of the air from the tips of the wing as shown in figs. 3,003 and 3,004.

Forces Acting on an Aeroplane.—When a machine is in flight there are four forces acting on it: 1. Lift (acting upward). 2. Gravity (acting downward). 3. Thrust (acting forward). 4. Air force (acting rearward).*

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*NOTE.—The author has avoided using the term *drift* for the air force because it is meaningless, confusing, and therefore objectionable. It would have been better if such terms as drift, keel surface, aspect ratio, etc., had never been coined.

For equilibrium there must be sufficient lift to balance gravity, and sufficient thrust to balance the air force. In other words, there are two pairs of forces, as shown in fig. 3,007, acting respectively vertically and in the lines of flight, and each pair are in balance—that is, the thrust or pull due to the propeller is equal to the force of the air acting on the machine.

The ratio of lift to this air force determines largely the value of a wing curve, and in general, that curve which gives the greatest amount of lift with the least air force at the speed for which the machine is designed is the most efficient.

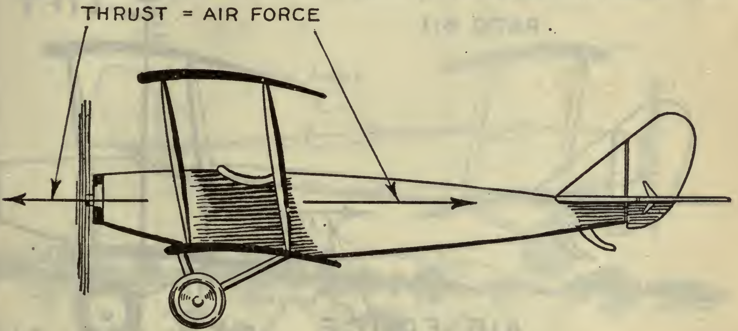


FIG. 3,008.—Relation between the thrust of the propeller and air force or resistance to the forward motion of the machine due to the air.

The ratio of lift to air force varies approximately from 1 to 1 on a flat plate to 8 or 10 to 1 for wings of special forms. The ratio 8 to 1 is easily obtained. This means that for every pound of forward thrust given by the propeller, 8 pounds of weight can be lifted vertically at the normal speed of the machines.

Center of Gravity.—By definition, the center of gravity of a body is *that point about which, if suspended, all the parts will be in equilibrium*.

The position of the center of gravity of an aeroplane has a marked influence on it's behavior in the air; it should, therefore, be known, and accordingly should be measured and marked on the machine.

The conditions would be ideal if all the forces acted at the center of gravity, but in construction the line of thrust and line of resistance does not usually coincide, because the line of thrust coincides with the propeller shaft axis, and this is usually above the line of resistance.

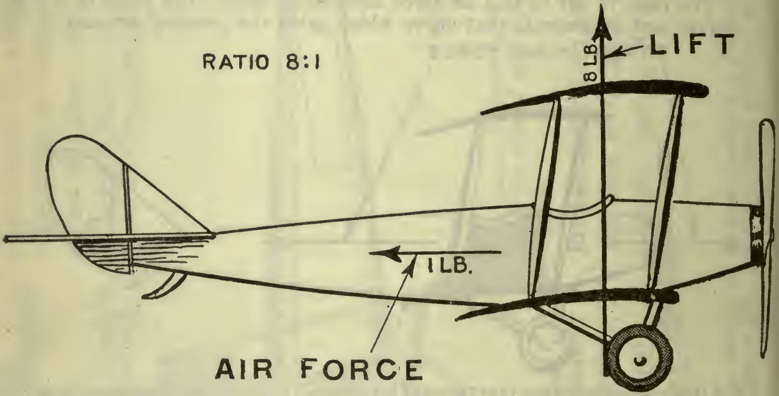


FIG. 3,009.—*The lift, air force ratio.* If for each 1 lb. of resistance (air force) overcome by the forward thrust of the propeller the upward pull or *lift* be, say 8 lbs., then the lift air force ratio is 8:1.

With the line of thrust above the line of resistance there is a tendency for the machine to nose down, and in order to balance this the center of gravity is located a sufficient distance behind the center of lift to give an equal tendency to tip the nose upward.

The position of the center of gravity affects lateral stability; if too low, it causes the machine to roll sidewise; if too high, it tends to upset the machine sidewise. The position of the center of gravity which gives maximum lateral stability is over a little below the center of air force.

Longitudinal Stability.—The angle of flight of an aeroplane with respect to the horizontal is controlled by the elevators. In flight, the center of lift changes its position whenever the angle of incidence or speed is changed.

If an aeroplane be flying at an angle of, say, 2° so that the center of gravity coincides with the center of lift for this angle, and a gust of wind cause the angle to increase ¼ degree, the center of lift will move forward and tend to elevate the front edge of the wing, thus further increasing the angle. The center of lift will then move further forward to accommodate the increase of angle, and the wing would rear up unless it were firmly attached to the aeroplane body and held in its proper position by the tail. Similarly if the angle be decreased, the tendency would be to depress the wing. It has been found that by giving the stabilizer a smaller angle of incidence than the wings, the disturbing force just described is opposed by a strong righting force.

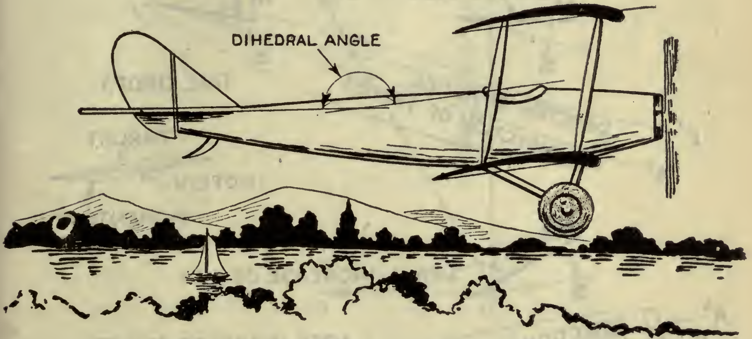
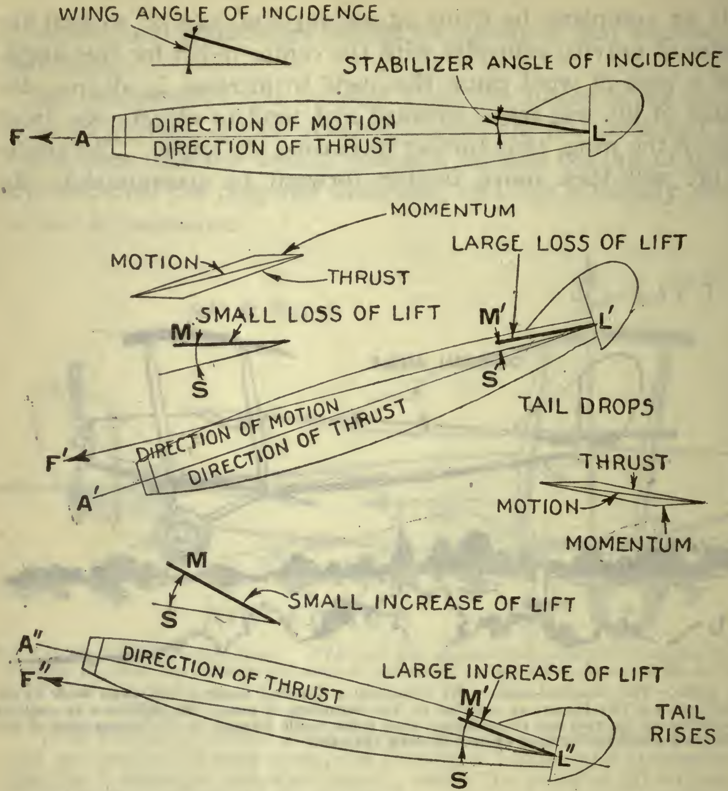


FIG. 3,010.—The dihedral angle. By definition a dihedral angle is *that angle made by the intersection of two planes*, as applied to the aeroplane it means *the difference in angle of inclination of the wing and the stabilizer*, that is the angle formed by the intersection of the wing and stabilizer surfaces projected along the axes.



Figs. 3,011 to 3,015.—Diagram showing righting effect of stabilizer and why stabilizer angle of incidence must be less than wing angle of incidence. ***In normal flight*** the direction of motion LF, and direction of thrust LA, coincide, as in fig. 3,011. Now, if a gust of wind throw the head down, as in fig 3,013, then the direction of motion will not coincide with the direction of thrust, but will take the direction of the resultant L′F′, of the thrust and motion, as found by the parallelogram fig. 3,012, and in fig. 3,013 it is seen that the stabilizer angle of incidence M′S′, is now much smaller in *proportion* than the wing angle of incidence MS, hence *the stabilizer has suffered a greater loss of lift than the wings, and the tail falls*, thus automatically bringing the machine back to its normal position, fig. 3,011. ***Again*** if a gust of wind cause the head to rise to position L″A″, as in fig. 3,015, there will result a small increase of wing angle of incidence and a relatively large increase of stabilizer angle of incidence, hence the stabilizer receives a greater increase of lift then the wings and the tail rises, thus automatically bringing the machine back to its normal position. Evidently the stabilizer must be fixed at a *less* angle of incidence than that of the wings, otherwise when the line of motion and line of thrust do not coincide, the two incidences would be altered in the *same* proportion and the righting effect would not be obtained. In the figures the wing and stabilizer are shown straight instead of curved for clearness.

A lack of longitudinal stability may be due to

1. Wrong stagger.
2. Warped body.
3. Wrong angle of incidence.

If there be not enough stagger, the lift will be too far back, tending to make the machine nose heavy.

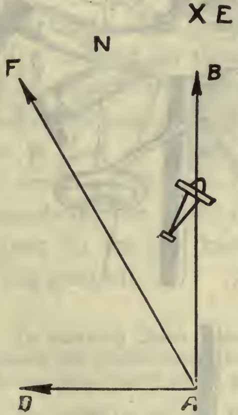
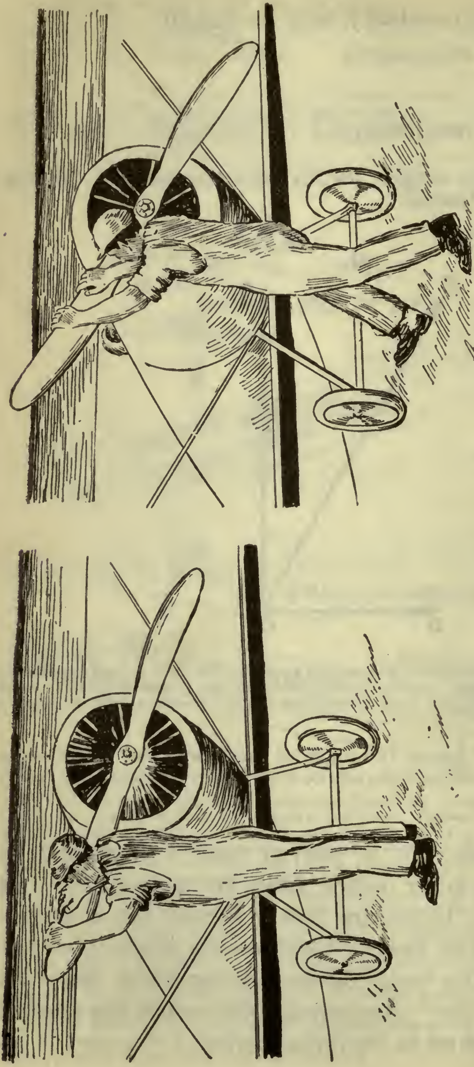


FIG. 3,016.—Diagram showing deviation of aeroplane to correct for drift caused by side winds. The angle at which the aeroplane must be pointed from the true course so that the resulting motion will be toward the objective point is obtained by the use of a drift set such as is shown in fig. 3,973.

A warped body may change the angle of incidence, and if the angle of incidence be too great, the machine will be nose heavy; if too little, it will be tail heavy.

Effect of Altitude.—Air is heavier near the ground than higher up, because the lower layers have to support the weight of the upper layers, hence the lower layers are compressed into smaller space. The heavier each particle of air, the greater is the inertia when the wing drives the particles downward, hence the greater the lift. Accordingly the nearer the aeroplane is to sea level, the greater is its lifting force.

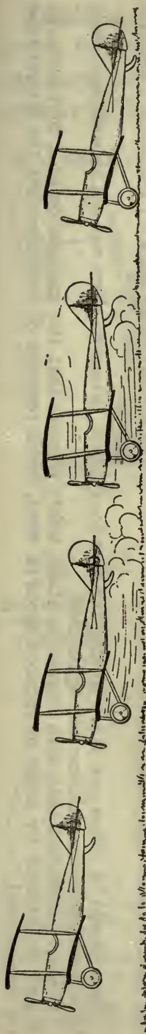


FIGS. 3,017 and 3,018.—Wrong and right methods of cranking. By gripping the blade near the middle instead at end, the operator can crank quicker and through a greater arc, and can also get out of the way quicker.

As the machine climbs to high altitudes, the density decreases and the speed increases, but due to the lighter air the engine takes in less charge of fuel mixture, hence the power is reduced, so, taking this into account, the speed is not so great in high altitudes as in low. It follows that aeroplanes to be used for flying in high altitudes must have greater wing area than those for low altitudes.

How to Fly

Preliminary Inspection.—The importance of inspecting every part of the machine before flight cannot be overestimated, because of the very light construction the safety of the operator depends on the perfect condition of the numerous delicate members which are subject to severe stresses. This includes attention to engine carburetter, ignition system, and maintaining a supply of gasoline, oil, and water.



FIGS. 3,019 to 3,022.—*Starting*. Fig. 3,022, machine at rest, fig. 3,021, accelerating, figs. 3,020 and 3,019, machine rising from the ground. In figs. 3,022 and 3,021, while moving with wheels on the ground the machine is said to be rolling.

Starting.—The machine should be turned around so that it faces the wind, for evidently the speed relation to the ground necessary to cause the machine to rise, is not so great as if started with the wind. This is because the lifting force depends on the speed of the machine with respect to the air, instead of with respect to the ground.

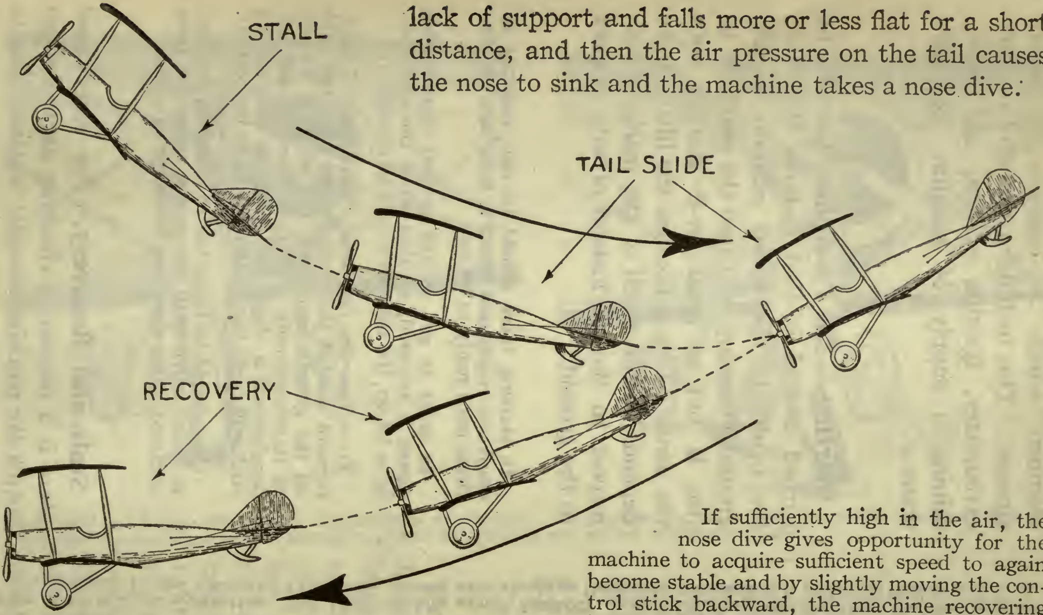
In starting there should be one assistant to crank the engine and others to hold back the machine until ready to let go.

In order to get off the ground considerable power is required, so that when the assistants have let go, the throttle should be opened.

To reduce the resistance, move control stick forward so that the elevator will raise tail skid off the ground and allow the machine to roll on its wheels only. When sufficient speed has been gained a slight pull back on the control stick will cause the machine to leave the ground.

Care should be taken not to cause the machine to climb much until the proper speed has been obtained, otherwise there is danger of *stalling*.

Stall and Recovery.—*For every speed there is a maximum climbing angle, beyond which the machine becomes unstable* due to lack of support and falls more or less flat for a short distance, and then the air pressure on the tail causes the nose to sink and the machine takes a nose dive.



Figs. 3,023 to 3,027.—Stall, tail slide and recovery.

If sufficiently high in the air, the nose dive gives opportunity for the machine to acquire sufficient speed to again become stable and by slightly moving the control stick backward, the machine recovering its normal flying angle by executing a graceful curve.

Care should be taken not to try to stop the dive too quickly, otherwise there will not be sufficient speed for stability and the machine will again stall.

It must be evident that with respect to stalls, safety requires that the machine be at a high altitude. A stall is detected by sense of reduced speed of wind, feeble response of controls—first the lateral then the longitudinal.

Gliding.—This is practically the same thing as "coasting" down hill in an automobile. The angle of incidence remains the same but the machine is pointed slightly downward so that gravity will supply the power instead of the engine.

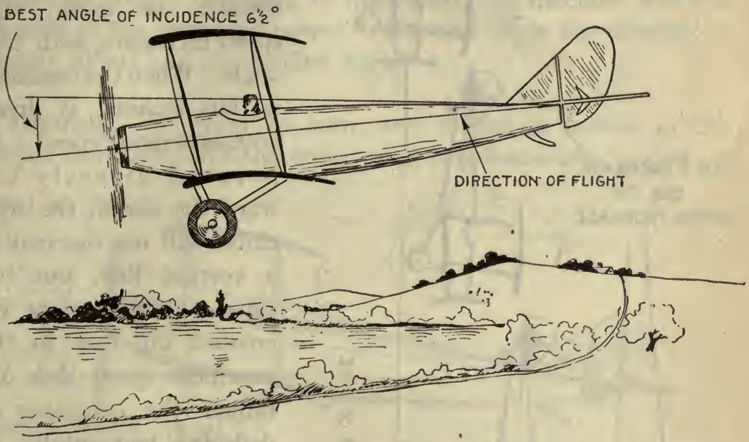
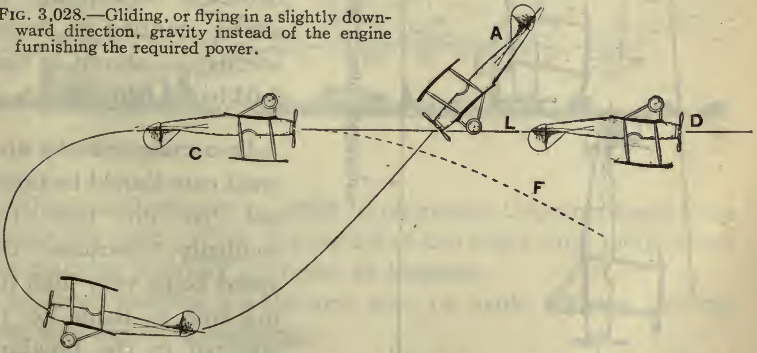
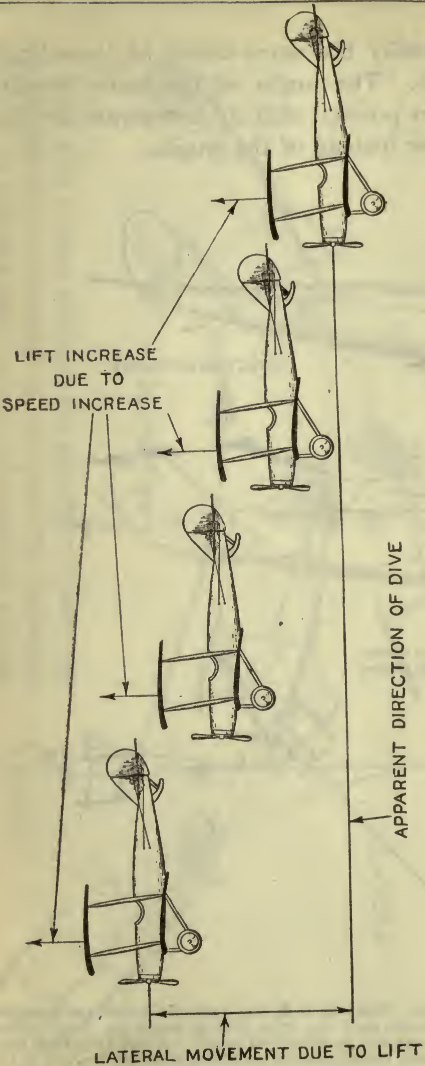


FIG. 3,028.—Gliding, or flying in a slightly downward direction, gravity instead of the engine furnishing the required power.



FIGS. 3,029 to 3.032.—Flying upside down. Starting at **A**, the operator points the machine downward to gain speed and then executes part of a wide loop to reach the upside down position **C**. Here the engine may or may not be throttled down. If not throttled, then the elevation must be operated to secure a course approximately in the direction **L**. If the engine be throttled, then a more inclined course **F**, must be taken.


LIFT INCREASE DUE TO SPEED INCREASE
APPARENT DIRECTION OF DIVE
LATERAL MOVEMENT DUE TO LIFT

Diving.—If the control be moved forward, the tail will rise, the machine inclining more and more toward the earth, the speed increasing with the angle. When the machine is thus inclined, it dives with enormous speed. If pointed directly toward the earth, the machine will not descend in a vertical line, but because of the greatly increased lift due to the enormous speed thus obtained the aeroplane is deflected horizontally or across, canting as it descends, as shown in figs. 3,033 to 3,036.

In coming out of a dive great care should be taken not to do this too suddenly, because the speed being very high the machine is liable to be strained to the breaking point.

Figs. 3,033 to 3,036. The dive, showing lateral motion of the machine due to increase of lift.

Climbing.—By slightly pulling back the control stick and holding it in a fixed position the machine will point upward and climb.

In some machines when flying horizontally if the speed of the engine be simply increased to maximum, the machine will rise because of the increased power. The best angle for climbing is usually about half the stalling angle.

Turning.—In order to turn, the rudder is moved in the proper direction, and the outer wing simultaneously tipped up by the lateral contact in order to overcome the centrifugal force due to the turn. This tipping up of the outer wing or inclining the machine laterally is known as *banking*.

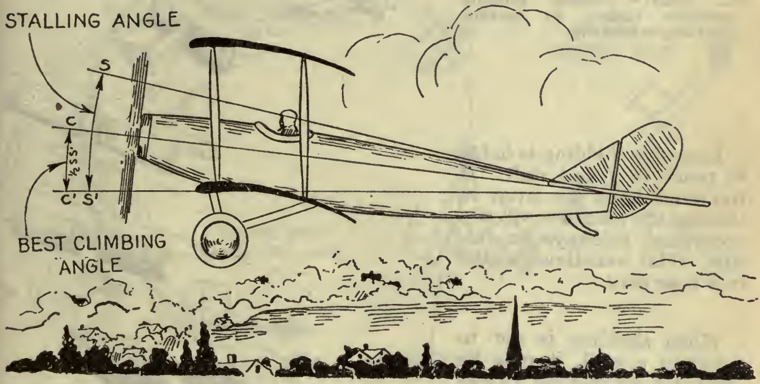


FIG. 3,037.—Best angle for climbing.

In calm weather a wide turn may be made without any appreciable degree of banking.

The proper bank for any particular turn is that bank which keeps the machine at the same angle of incidence relative to the air without rising or falling.

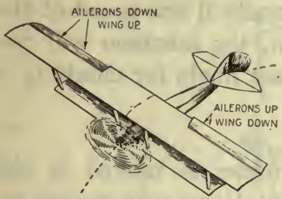


FIG. 3,038.—Turning, showing machine banked to prevent skidding or side slip.

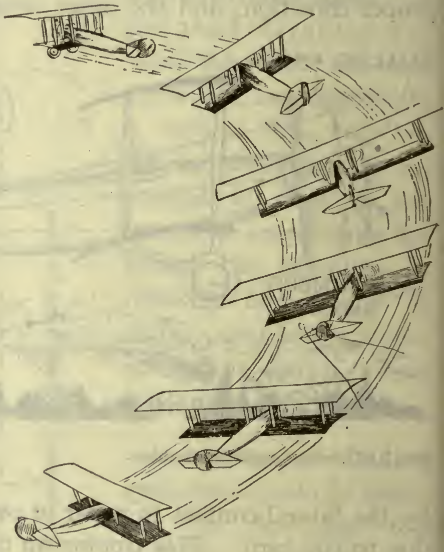
When a turn is properly made the operator does not feel any change in wind pressure or tendency of the body to swing sideways but only a slight increased pressure in the seat.

If the machine be not properly banked in making a turn it will skid outward, which is noticeable by the side wind thus created.

Operators sometimes skid considerably in turning in order to quickly reduce the speed of the machine just before landing.

Excessive skidding is liable to result in a bad stall. If too much bank be given in turning the machine will slip downward, sideways or *side slip*. This sometimes results in a nose dive.

When skidding is apt to result in a stall, it may be overcome by moving rudder back toward neutral or by increasing the bank.



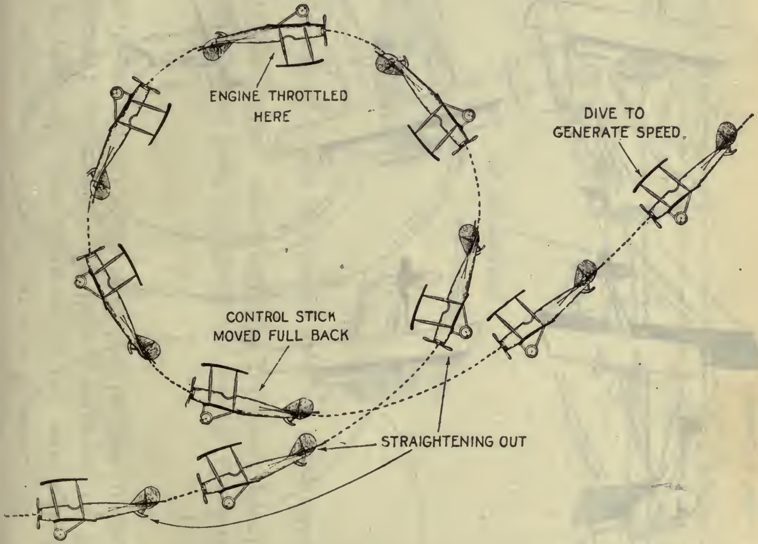
FIGS. 3,039 to 3,044.—Simultaneous climbing and turning

If there be danger of a nose dive by side slipping, this is first over.

In climbing while making a turn care should be taken to sufficiently bank the machine and not to give too much rudder, otherwise there is danger of a stall.

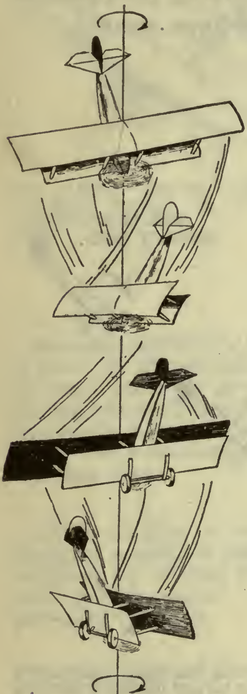
If the machine stall under these conditions the rudder should be put over to the opposite side and the bank also changed to opposite side.

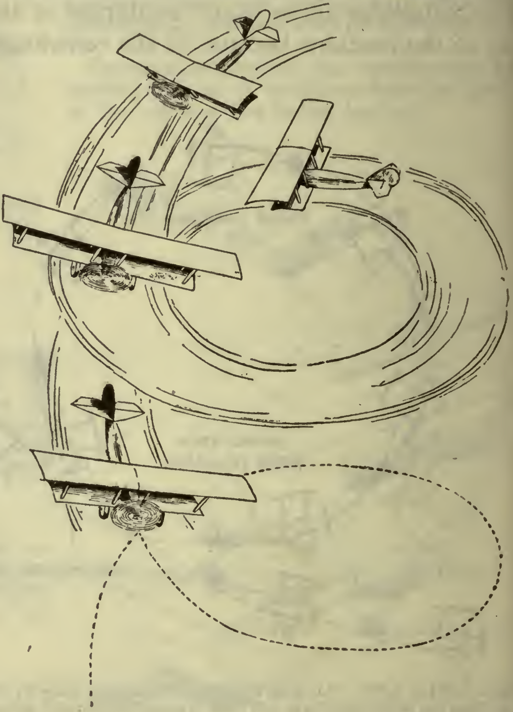
Looping.—This is done with the operator on the inside of the loop, otherwise there would be danger of throwing the operator out of the machine because of the centrifugal force created.



FIGS. 3,045 to 3,054.—Looping the loop. The loop is made by diving first inside circle method so that the centrifugal force will tend to press the operator against his seat instead of throwing him out of the machine.

To loop the loop: 1, make a semi-dive to develop the required speed; 2, pull elevator all the way back with engine going full speed, which quickly points the machine upward; 3, when the machine reaches the top of the loop, throttle engine, allowing momentum to carry the machine over; 4, as the machine speeds up in completing the loop, the control stick should be eased back toward neutral so that the machine will not straighten out to normal flying angle too quickly and bring undue stresses on the wings. As the machine begins the descent to complete the loop it should be prevented going over sideways by means of the rudder.





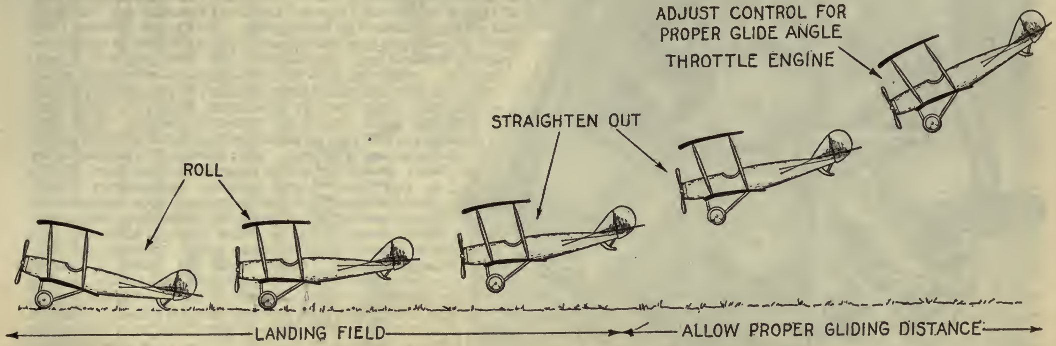
FIGS. 3,055 to 3,058.—Nose spin dive, so called because the axis of rotation is through the nose of the machine.

FIGS. 3,059 to 3,062.—Spiral or cork screw dive.

Nose Spinning.—This consists of a rotation of the machine (in descending) on an axis passing through the nose but at an angle with the body of the machine. Some machines are subject to this after a stall with the engine running. To come out of a spin, bring all controls to neutral and hold them there, then dive far enough to regain control.

Landing.—The machine should approach the spot selected for landing, from such distance that it will travel toward it with the proper gliding angle.

To descend, throttle the engine and push control stick slightly forward to bring the machine to the proper gliding angle.



FIGS. 3,063 to 3,067. Proper method of landing.

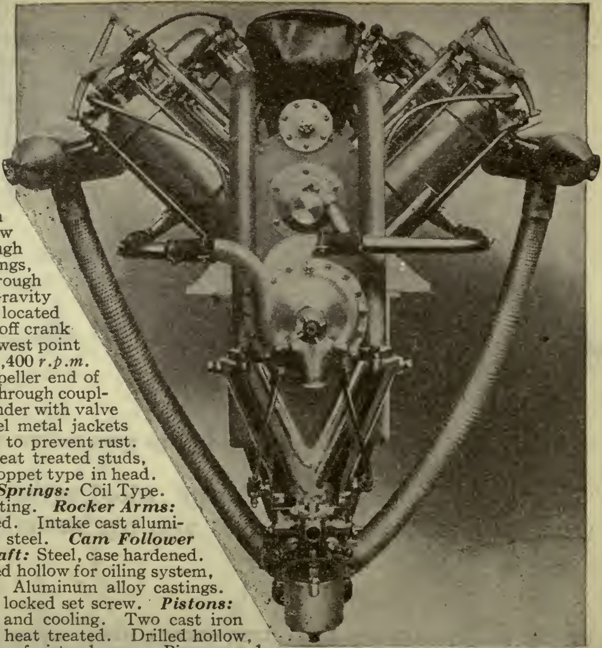
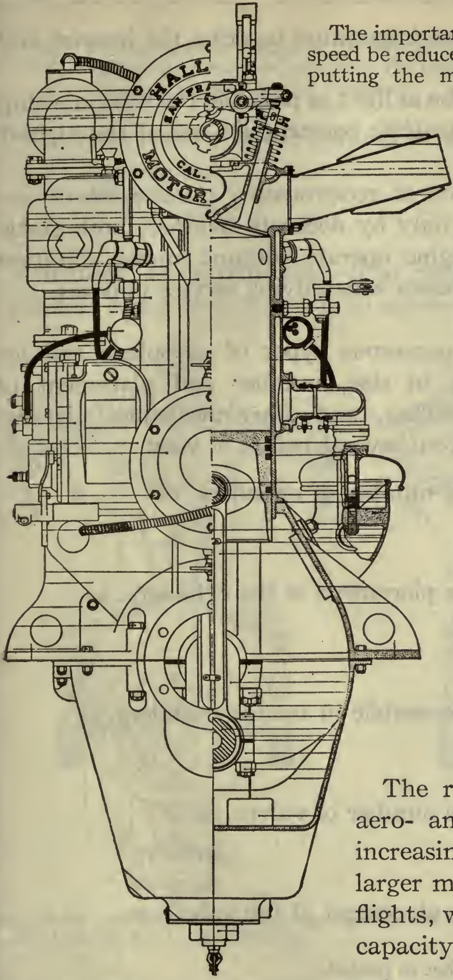


FIG. 3,068.—Curtis eight cylinder V engine. ***Rated horse power*** at 1,400 *r.p.m.*, 90; ***cylinders,*** 4×5. ***Ignition:*** One high tension, single spark, 8 cylinder magneto located at top of crank case and driven off timing gears at front. Leather boot on magneto. Rubber covers on spark plugs. Spark plugs metric thread 18 mm., 1½ mm., pitch. ***Carburetter:*** "Zenith" duplex. Each intake supplies one bank of four cylinders through separate manifolds. Lower end of manifolds water jacketed. Auxiliary air intake in manifolds with hand control for altitude adjustment. Carburetter located under front end of engine, and braced by steel struts to crank case. ***Lubrication:*** Pressure feed. The oil is forced from lowest point of sump by gear pump to rear end of hollow cam shaft, through pressure adjusting valve through cam shaft to timing gears and all cam shaft bearings, thence through tubes to crank shaft bearings and through crank shaft throws to connecting rod bearings. Gravity return to splash pan, and oil reservoir. Oil pump is located at bottom of propeller end of crank case and is driven off crank shaft through bevel gear. Removable oil screen at lowest point of sump. Oil pressure from forty to sixty pounds at 1,400 *r.p.m.* ***Water Circulation:*** Centrifugal pump bolted to propeller end of crank case and driven direct off end of crank shaft through coupling. ***Cylinders:*** Each casting embodying one cylinder with valve chambers and valve stem guides cast integral. Monel metal jackets brazed on. Outside of cylinder heavily nickel-plated to prevent rust. Cylinders bolted to crank case by eight nickel steel, heat treated studs, four of which extend to top of cylinder. ***Valves:*** Poppet type in head. Intake, nickel steel. Exhaust, tungsten steel. ***Valve Springs:*** Coil Type. ***Valve Cages:*** Valves seat direct in cylinder head casting. ***Rocker Arms:*** Exhaust, drop forgings, case hardened and nickel plated. Intake cast aluminum. ***Cam Followers:*** Plunger type, case hardened steel. ***Cam Follower Guides:*** Manganese bronze, bolted to case. ***Cam Shaft:*** Steel, case hardened. Eight double acting cams integral. Five bearings drilled hollow for oiling system, bearings and cams ground. ***Cam Shaft Bearings:*** Aluminum alloy castings. Split type bolted together and held in crank case by locked set screw. ***Pistons:*** Aluminum alloy castings with deep ribs for strength and cooling. Two cast iron eccentric rings. ***Piston Pins:*** Chrome nickel steel, heat treated. Drilled hollow, case hardened and ground. Pin bearings in aluminum of piston bosses. Pins pressed in connecting rods and held by locked set screw. ***Connecting Rods:*** Chrome Vanadium steel forgings, "H" section machined all over and heat treated. ***Connecting Rod Bearings:*** Bronze backed, babbitt lined. Reamed for crank shaft. Held in rod by four brass rivets. Cap bolted on with two bolts. ***Crank Shaft:*** Heat treated chrome nickel steel forging, finished all over and drilled hollow. Four throw, five main bearings. All bearings ground. End of shaft splined, tapered and threaded for propeller hub. ***Crank Shaft Bearings.*** Bronze backed, babbitt lined. Line reamed to crank shaft. End caps aluminum, others steel. All caps held on with two large bolts.



The important thing in landing is that the speed be reduced as much as possible without putting the machine in a climbing angle, therefore in landing the machine should be facing the wind.

When within say 14 or 20 feet of the ground the machine should be straightened out to horizontal flying position and allowed to gradually sink to the ground, being careful to keep the wings from tilting sideways by means of the horizontal control.

In *rolling* after the wheels touch the ground, keep the tail as close to the ground as possible without causing bumping so as to slow up the machine as quickly as possible.

One bad form of landing is called the *pancake*, which results from reducing the speed too much by pointing the machine upward to climbing position. This causes a stall and the machine drops to the ground tail first.

Aeroplane Engines

The rapid development of the aero- and sea-plane is gradually increasing, and the demand for larger machines, capable of long flights, with greater load carrying capacity, more powerful and

FIG. 3,069.—Front end sectional view of Hall-Scott 100 horse power four cylinder vertical engine. *R.p.m.* at which horse power is rated, 1,400; recommended *r.p.m.*, 1,300; minimum gasoline consumption at 1,300 *r.p.m.*, 10 gallons; oil per hour, ½ gallon; maximum radiating surface for front type radiator, 400 sq. ins.

highly efficient engines, are imperative to drive the heavier and speedier machines.

Since the engines must be as light as possible, the workmanship and materials entering into their construction are of the highest order.

The question of balance of reciprocating parts receives unusual attention, because only by accurate balance, both static and running, can the engine operate without undue vibration and endure the severe stresses which flying service imposes.

Classification.—The numerous types of aeroplane call for various engines differing in size, number and placement of cylinders, methods of cooling, and other mechanical details. These may be classified from several points of view.

1. With respect to the number of cylinders, as
   - *a.* Four cylinder.
   - *b.* Six cylinder.
2. With respect to the placement of the cylinders, as
   - *a.* Vertical.
   - *b.* Inclined (V type).
   - *c.* Radial.
3. With respect to non-mobile or mobile cylinders, as
   - *a.* Stationary.
   - *b.* Rotating.
4. With respect to the number of valves, as
   - *a.* Single.
   - *b.* Double.
5. With respect to the placement of the valves, as
   - *a.* Valves in head.
   - *b.* One valve in head, other in pocket.

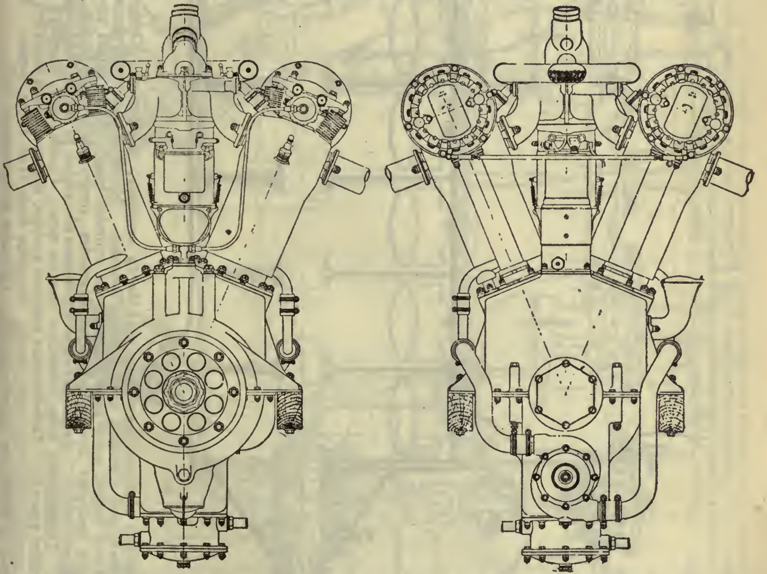
NOTE.—In the treatment of aeroplane engines, the author assumes that the reader has a general knowledge of gas engine principles and operation, because otherwise there would be a useless repetition of matter that has already been presented in the chapter on gas engines.

c. Valves on same side, side by side (L head).
d. Valves on same side, one over the other.
e. Valves on opposite sides (T head).

6. With respect to the method of cooling, as

a. Water cooled.
b. Air cooled.
c. Semi-air cooled.

Number of Cylinders.—The multiplicity of gas engine cylinders is due to a desire, 1, to secure a nearer uniform turning torque, and 2, to reduce the vibration. For aeroplanes of small and moderate power 6 and 8 cylinder engines are standard, and for high power, 12 or more cylinders are used, as it must be



Figs. 3,070 and 3,071.—Transverse elevations of ***Liberty 12 engine.*** Fig. 3,070, front elevation, fig. 3,071, rear elevation.

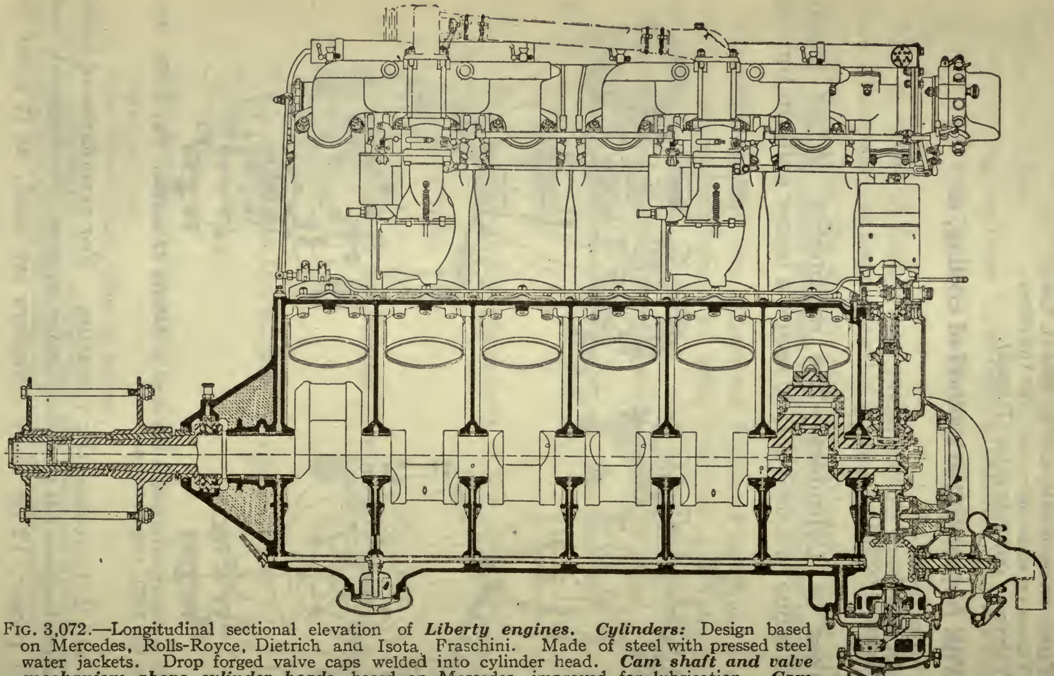
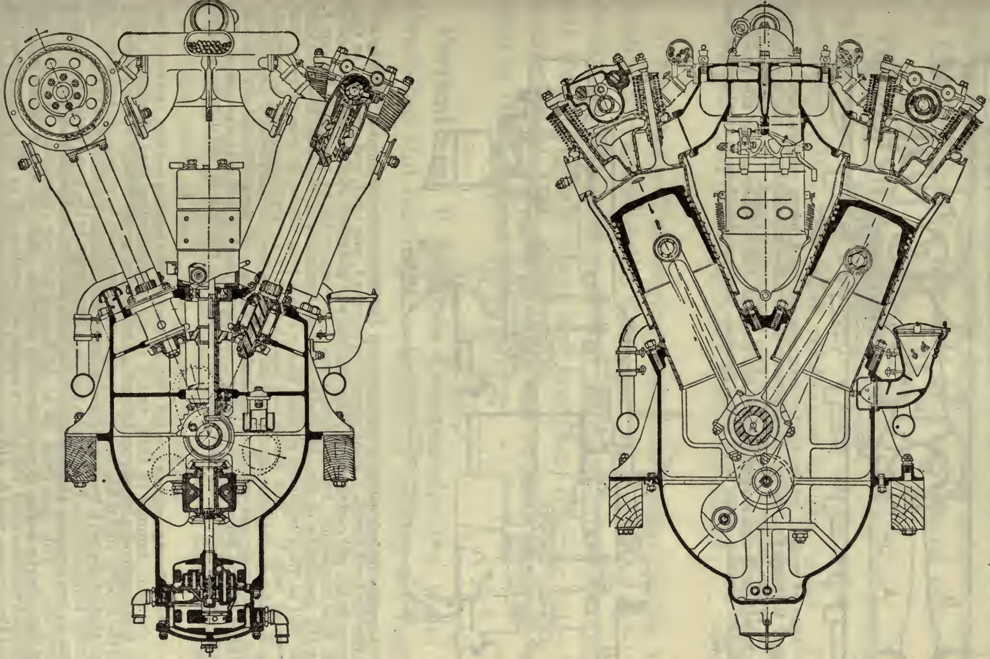


FIG. 3,072.—Longitudinal sectional elevation of ***Liberty engines.*** ***Cylinders:*** Design based on Mercedes, Rolls-Royce, Dietrich and Isota Fraschini. Made of steel with pressed steel water jackets. Drop forged valve caps welded into cylinder head. ***Cam shaft and valve mechanism above cylinder heads,*** based on Mercedes, improved for lubrication. ***Cam shaft drive,*** Hall-Scott design. ***Angle between Cylinders*** 45(, object to reduce head resistance. ***Ignition,*** Delco. ***Pistons,*** Hall-Scott design. ***Connecting rods,*** forked or straddle type. Crank shaft, standard 12 cylinder practice except to oiling. ***Crank case,*** standard practise. ***Lubrication*** pressure system, the crank pins being fed through a hole inside the crank cheek instead of up the outside face of the crank cheek. ***Propller hub,*** Hall Scott design. ***Water pump,*** Packard type. ***Carburetter,*** developed by Zenith for Liberty engine. ***Cylinder dimensions,*** 5×7.



FIGS. 3,072*a* and 3,072*b*.—Transverse sectional elevations of ***Liberty engine.*** Fig. 3,072*a*, rear elevation, showing turning gear train; fig. 3,072*b*, elevation through cylinders, showing valves, pistons, connecting rods, etc. Crank shaft main bearing journals, 2⅝ diam.; crank pins, 2½. Connecting rod ratio 1.71. Pistons die cast of aluminum alloy, tapered walls to efficiently transfer heat. Piston pin floating type held by aluminum washers pressed into the piston at each side. Valves 2½ diam.

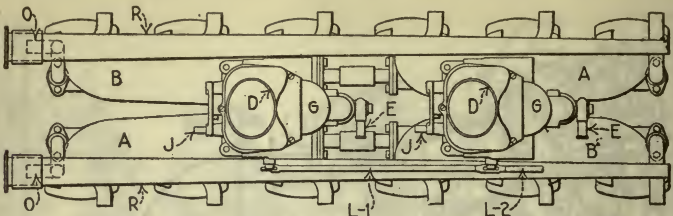


FIG. 3,072*c*.—Plan of revised intake header arrangement on ***Liberty engine,*** and the way it is necessary to bend the water outlet pipe to permit the installation of the new intake pipe B. The intake header A, is provided with nipples G, at each end so that overflow pipes may be attached and carried outside the fuselage. It is obvious that this provided for the complete draining away of all stray gasoline, whether the plane is climbing, diving or flying level.

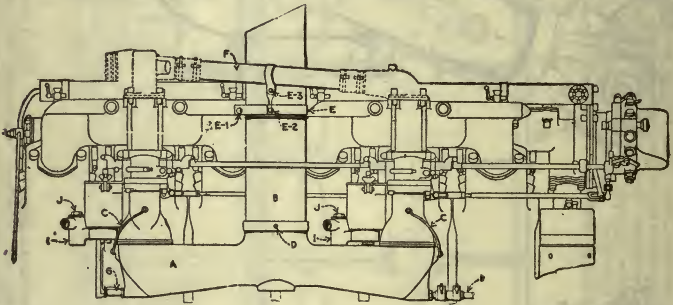


FIG. 3,072*d*.—Longitudinal elevation showing revised carburetter arrangement in ***Liberty engine.*** To provide for a screen, a small aluminum casting 1, was used, and arranged so that it would screw on the old carburetter. This casting was provided with a nipple to receive the gasoline hose connection, and the screen was easily removable by taking off the nut J. To prevent the possibility of fire, the individual air intakes were removed from the carburetter, as well as their retaining springs, and a single long intake A, was provided and held on by spring bails C. This intake was machined at its center to receive a large single intake pipe B, extending up through the center of the engine and out through the bonnet. The top end of this pipe was cut off at an angle of 25 deg. A dowel D, engaged a slot in the intake pipe B, and prevented it turning. A clamp band E and E1, was arranged to anchor the intake pipe B, to suitable cap screws already existing in the intake headers. shown in fig. 3,072*d*.

NOTE.—"The ***Liberty engine*** revised altitude control was worked out first and went into production in the early summer of 1918, and it is my understanding that all engines were equipped with this improved altitude control before being used at the front. Vague rumors to the effect that the Liberty engine has excessive gasoline consumption have been pretty generally circulated, and this, of course, if true, would be serious. It is obvious that all other things being equal, the gasoline consumption of the engine will be in proportion to the horse power developed. All well-known air craft engines at the present time have about the same gasoline consumption per horse power hour. This runs from .48 to .56 lb. per *b.hp.-hr.*, depending on conditions. In this respect the Liberty engine is no better, and no worse, than other well-known designs. Running wide open near sea level, the Liberty engine will develop 400 to 420 *hp*. and granting that it develops the average, that is, 410 *hp*., and that the average service conditions maintain, it will probably have a gasoline consumption of .52 lb. per *b.hp.-hr.*"—*Vincent.*

evident that in order to keep down the cylinder dimensions to proper limits so as to avoid destructive vibration, the number of cylinders must be increased with increase of power required.

Vertical Cylinder Engines.—In this type of engine all the cylinders are placed in line, one back of the other, and it accordingly offers the least resistance to the wind. The long crank case, while not so light as some types, permit bearings of liberal length, thus insuring cool running and durability. The valve gear and other parts are easily accessible.

V Engines.—This engine lends itself conveniently to water cooling and although the balance of the prevailing eight cylinder design is not so good as that of the multi-cylinder radial engine, yet the V engine is short and compact, offering little head resistance and view obstruction, and appears likely to be much used in the future of air service.

Vertical and V Engines Compared.—For equal cylinder dimensions and number of cylinders, V engines are lighter than the vertical, but in actual practice, for equal power output, the V type has a greater number of cylinders, in order to secure nearer uniform torque by increasing the number of impulses.

Since reducing the size of cylinder increases weight per horse power this is offset by running the engine at higher speed than the vertical type. However a strong reduction gearing for the propeller drive is required, and since there is loss of power in this gearing there is not much saving in weight per horse power of the V over the vertical engine.

Radial Engines.—The fear, on the part of the designers, of the evils resulting from excessive cylinder lubrication resulted n the early radial engines exhibiting a fan like arrangement of the cylinders as in fig. 3,037, the arcs of all being inclined upward from the crank shaft.

The fan type has not survived, increased experience having shown that with carefully designed forced lubrication the cylinders may be symmetrically disposed around the crank shaft without fear of those below the horizontal becoming flooded.

An advantage of the radial engine with cylinders placed all around the crank case is the considerable reduction in weight and simplicity of the crank case, there being no intermediate bearings and webs.

The impulses, however, do not occur at equal intervals as with the more usual types, except in the case of a five cylinder engine. In construction the crank case is stationary and the crank shaft revolves, as distinguished from the so called rotary type in which the opposite conditions obtain.

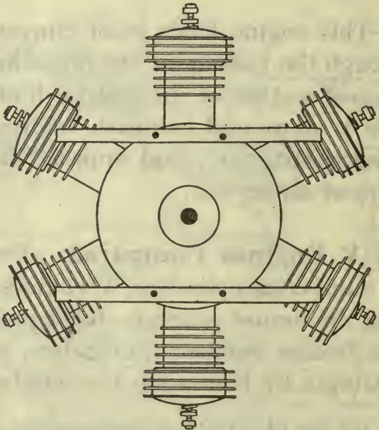


Fig. 3,073.—Six cylinder air cooled radial engine. ***It has*** fixed cylinders arranged radially about a rotating crank shaft. It is the type built by Anzani.

The valves are placed in the cylinder head. The large surface offers too much resistance to the wind, and as a rule the engine has to be placed in front of the operator.

Rotating or So Called Rotary Engines.—In general, rotating engines resemble in external appearance the radial type, but, whereas in radial engines, the cylinders are stationary and the crank shaft rotates, in rotating engines the crank shaft is fixed and the cylinder, crank case, and attachments rotate around the crank shaft axis.

While the rotary engine has the advantage of lightness, and the steadying influence of its large rotational inertia, it has certain drawbacks: it must necessarily be air cooled and for long flights water cooling is preferable.

Again the resistance of the air to the rapid rotation of the cylinders absorbs fully 10% of the power developed while it is difficult to provide for the uniform cooling of the cylinders and so avoid distortion, the leading surfaces tending to keep cooler than the following surfaces.

It has proved difficult to arrange satisfactorily for the supply of the carburetted mixture to the several cylinders, and there is no satisfactory way to muffle the exhaust.

Rotating engines have an odd number of cylinders in order to secure evenly spaced explosions.

The consumption of lubricating oil is excessive because of centrifugal force.

The Gnome Rotating Engine.—This engine is erroneously called a "one valve" engine, because there is only one valve of the ordinary type in each cylinder—the exhaust valve, which is located in the top of the cylinder head.

The other valve for inlet is formed by the piston which as it nears the end of the outward (away from the head) stroke uncovers a ring of small holes drilled through the cylinder and communicating with the crank case, the idea being similar to that employed in the uniflow type steam engine, differing in that the ports uncovered are inlet instead of exhaust parts. This combination of piston and circumferential ports is virtually a valve and it is therefore ridiculous to speak of the engine as a one-valve engine.

In operation a very rich fuel mixture is forced into the crank case through a jet inside the crank shaft, and enters the cylinder when the piston is at the end of its outward stroke; that is, when the inlet parts are uncovered.

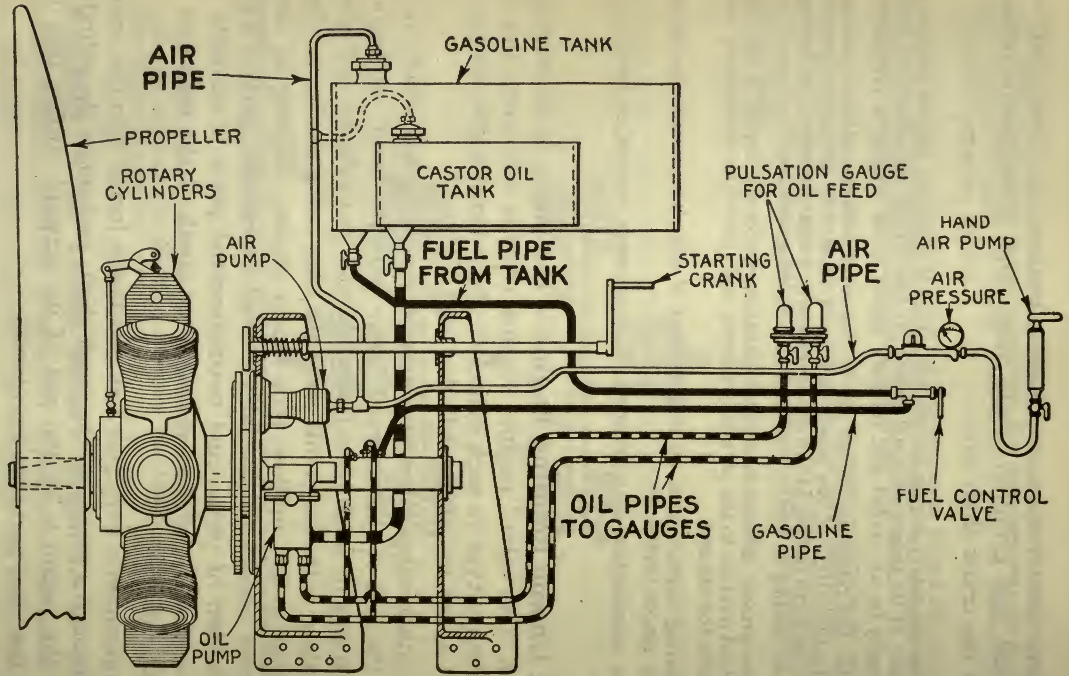
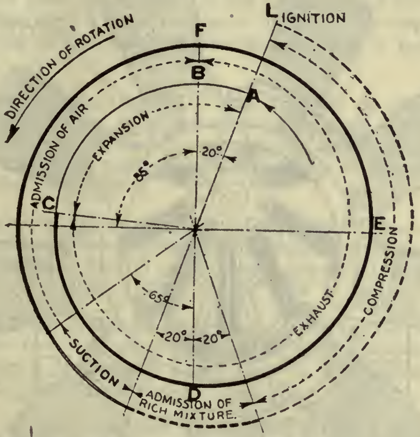


FIG. 3,074.—Piping diagram of rotating engine showing fuel, oil, and air lines, valves and gauges.

On the return stroke these parts are covered and the charge is compressed and ignited in the usual way, exhaust taking place through the large valve in the head. The exhaust valve remains open a portion of the intake stroke to admit air into the cylinder and dilute the rich gas forced in from the crank case.

The fuel supply, ignition and oiling of the Gnome engine has been described in "The Automobile" as follows:



IG. 3.075.—Timing diagram of Gnome engine. ***To time engine,*** it should be mounted on supports with front nose extension removed, together with the front cover with its ball rod and the two pairs of reducing or satellite gears. Turn engine anti-clockwise and adjust valve rod cup ends for .5 millimeter clearance between exhaust valve levers roller and exhaust valve of each cylinder when valve is closed. Then turn engine anti-clockwise until cylinder No. 1 is 85 degrees past top dead center. Turn cam shaft clockwise till No. 1 cam is about to lift valves. Take front cover with satellite pinions and turn satellite wheels till the two teeth marked O, are opposite each other. Fit front cover on valve gear case so that gears mesh and bolt holes are in line. *Verify setting.* Put on nuts and fit ball rod in position. Verify entire setting and adjust tappel clearance again if necessary.

Gasoline is fed to the engine by means of air pressure at 5 pounds per sq. in., which is produced by the air pump on the engine. A pressure gauge, convenient to the operator, indicates this pressure and a valve enables the operator to control it. No carburetter is used.

The gasoline flows from the tank through a shut off valve near the operator and through a tube leading through the hollow crank shaft to a spray nozzle located in the crank case.

There is no throttle valve, and as each cylinder always receives the same amount of air as long as the atmospheric pressure is the same, the output cannot be varied by reducing the fuel supply, except within narrow limits.

The high tension magnetos, with double cam or two break per revolution interrupter, is located on the thrust plate in an inverted position, and is driven at such a speed as to produce nine sparks for every two revolutions; that is, at 2¼ times engine speed. A Splitdorf magneto is fitted. There is no distributer on the magneto.

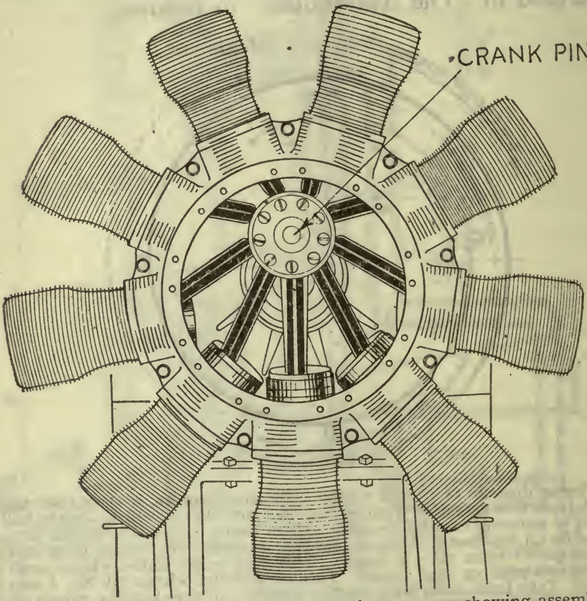


FIG. 3,076.—End view of Gnome engine with crank case open showing assembly of pistons and connecting rods.

The high tension collector brush of the magneto is connected to a distributer brush holder carried in the bearer plate of the engine.

The brush in this brush holder is pressed against a distributer ring o insulating material molded in position in the web of a gear wheel keyed t the thrust plate, which gear serves also for starting the engine by hand

Molded in this ring of insulating material are nine brass contact sectors connecting with contact screws at the back side of the gear, from which bar wires connect to the spark plugs.

The distributor revolves at engine speed, instead of at half engine speed as on ordinary engines, and the distributor brush is brought into electrical connection with each spark plug every time the piston in the cylinder in which this spark plug is located approaches the outer dead center. However, on the exhaust stroke no spark is being generated in the magneto, hence none is produced at the spark plug.

Ordinarily, the engine is started by turning on the propeller, but for emergency purposes as in sea planes or for a quick "get away" if landing inadvertently in enemy territory, a hand starting crank is provided. This is supported in bearings secured to the pressed steel carriers of the engine and is provided with a universal joint between the two supports so as to prevent binding of the crank in the bearings due to possible distortion of the supports.

The gear on this starting crank and the one on the thrust plate with which it meshes are cut with helical teeth of such hand that the starting pinion is thrown out of mesh as soon as the engine picks up its cycle. A coiled spring surrounds part of the shaft of the starting crank and holds it out of gear when not in use.

Lubricating oil is carried in a tank of 25 gallon capacity, and if this tank has to be placed in a low position it is connected with the air pressure line, so that the suction of the oil pump is not depended upon to get the oil to the pump.

From the bottom of the oil tank a pipe leads to the pump inlet. There are two outlets from the pump, each entering the hollow crank shaft, and there is a branch from each outlet pipe to a circulation indicator convenient to the operator.

One of the oil leads feeds to the housing in the thrust plate containing the two rear ball bearings, and the other lead feeds through the crank pin to the cams, as already explained.

Owing to the effect of centrifugal force and the fact that the oil is not used over again, the oil consumption of a revolving cylinder engine is considerably higher than that of a stationary cylinder engine. Fuel consumption is also somewhat higher, and for this reason the revolving cylinder engine is not so well suited for types of aeroplanes designed for long trips, as the increased weight of supplies required for such trips, as compared with stationary cylinder type engines, more than offsets the high weight efficiency of the engine itself, but for short trips, and especially where high speed is required, as in single seated scout and battle planes, the revolving cylinder engine has the advantage.

The oil consumption of the Gnome engine is as high as 2.4 gallons per hour. Castor oil is used for lubrication because it is not cut by the gasoline mist present in the engine interior as an oil of mineral derivation would be.

Cooling Methods.—There are two methods of cooling aeroplane engine cylinders, being similar to those employed in automobile practice.

1. Air cooling.
2. Water cooling.

In the most approved air cooling system provisions are made to direct the air to the cooling surfaces of each cylinder so that each receives an equal cooling effect. This is done by placing around each cylinder a jacket of thin sheet metal connected at each end with an air tight manifold.

In operation air is discharged into one of these manifolds, from whence it passes around the radiating fins of each cylinder, being guided by the sheet metal jackets, and is discharged through the other manifold.

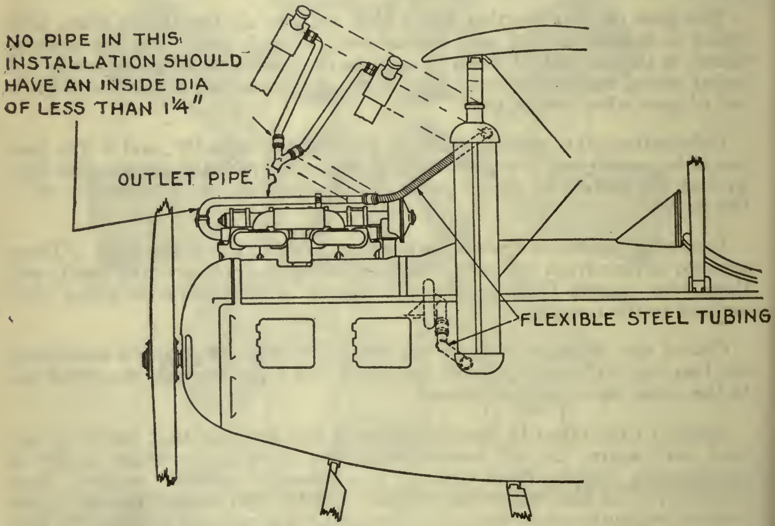

NO PIPE IN THIS INSTALLATION SHOULD HAVE AN INSIDE DIA OF LESS THAN 1¼"
OUTLET PIPE
FLEXIBLE STEEL TUBING

FIG. 3,077.—Hall-Scott side radiator installation for 100 horse power engine.

It is stated that the air system is more economical than water cooling because of the higher temperature maintained. However, the water system is conceded more reliable and is the one generally used. In this system a circulation of water is maintained between the cylinder jackets and the radiator by a rotary pump, or equivalent, the construction being similar to that used in automobile practice.

Carburetter Requirements.—Atmospheric conditions have much to do with carburetter operation, and especially in aeroplane service are these conditions varied to a marked degree.

Anyone familiar with gas engine operation has noticed the difference in power developed on a cold and on a hot day. This is due to the fact that the volume of air varies with its temperature, hence, when it is cold, it occupies a relatively small volume, and the engine takes in a larger "charge" than when the air is hot, hence greater power is developed.

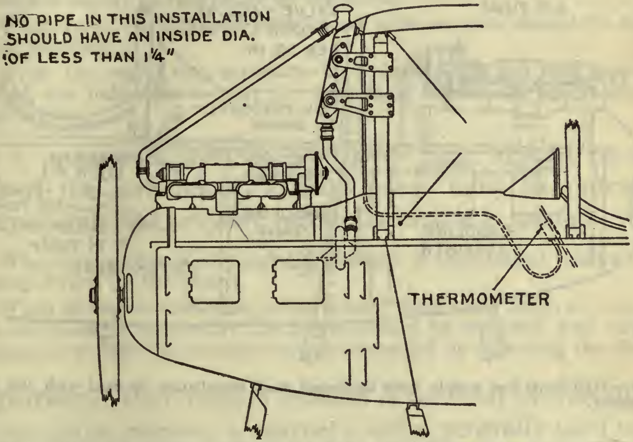

NO PIPE IN THIS INSTALLATION SHOULD HAVE AN INSIDE DIA. OF LESS THAN 1¼"
THERMOMETER

Fig. 3,078.—Hall-Scott overhead radiator installation for 100 horse power engine.

Under these conditions the mixture will become too rich unless the carburetter be provided with adjustments to maintain the mixture in the correct proportions. This may be done by providing minute adjustment of the gasoline by hand control, or else a hand control supplementary air valve located between the engine and carburetter. Both of these adjustments should be provided.

The volume or density of air also varies with the altitude, becoming lighter as the altitude increases. Accordingly, an aeroplane in ascending from sea level to a high altitude, takes in charges of air decreasing in weight.

The effect of the supplementary valve is to reduce the suction of the primary and secondary air supply so that there is less pressure reduction when the engine draws in the air, hence an increased amount of air will be taken in during each suction stroke.

Both of the adjustments should be provided on every aeroplane with hand control for the instrument board within easy reach of the operator.

The supplementary air valve should be provided, not only for altitude correction, but for economy, so that the operator can control the richness of the mixture, and be sure that the engine is running on the leanest mixture possible.

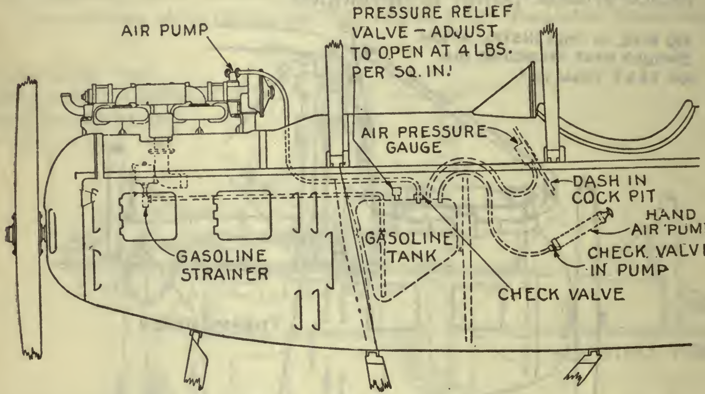


FIG. 3,079.—Hall-Scott fuel supply with depressed tank installation as used with 100 horse power engine.

Since an aeroplane is usually in an inclined position carburetters with offset float chambers should not be used because, due to such inclination the float level with respect to the nozzle is disturbed and may be higher, which would cause gasoline to leak out of the carburetter while standing with resulting danger of fire.

To guard against such danger, not only should a concentric float chamber be used, but provision should be made so that any gasoline finding its way into the primary air passage will drain overboard and not saturate the magneto or woodwork of the machine.

Fuel Supply.—There are two methods by which the fuel is brought from the tank to the carburetter, as by

1. Gravity, or
2. Pressure.

When using the gravity system there should be a drop of not less than two feet from the lowest part of the tank to the upper part of the carburetter float chamber.

Even this height might not be sufficient to maintain the proper volume of gasoline to the carburetter at high speeds, or when climbing at a considerable angle.

A combination of gravity and pressure should be employed to insure an adequate supply of fuel under all conditions.

When using gravity feed without air pressure care should be taken to vent the tank.

When the engine runs satisfactorily on low speeds and cuts out on high speed, the indication is that the tank is of insufficient height above the carburetter.

If a pressure system be used with tank located below the level of the carburetter, a hand pump must be provided to generate sufficient pressure before the engine is started.

When the engine is running, the pressure is maintained preferably by a pump driven by the engine.

When an engine driven air pump is used there must be an air relief valve on the tank, or preferably the pump should be designed with adjustable clearance so that the pressure can be controlled by adjusting the clearance.

Ignition.—High tension ignition is the kind universally used for aeroplane engines, a magneto being generally used in preference to a battery.

Since the engine causes practically all the hum at high speed and heavy load, the ignition apparatus must work under severe conditions. An extra high pressure is required to produce a spark because of the high compression due to wide throttle opening. Moreover, special provision must be made to prevent the plugs becoming unduly heated owing to the very high speed. It is of vital importance that the ignition system be maintained in perfect order, because the safety of the machine and occupant depends upon its proper working.

To guard against failure, the best practice is to provide two independent systems.

An example of a double ignition system is that fitted on the Sturtevant eight cylinder V engine.

It consists of two water proof magnetos placed face to face between the two groups of cylinders. Each cylinder is provided with double ignition by means of two spark plugs located in water cooled bosses on the sides of the cylinder heads. The magnetos are synchronized by a vernier coupling permitting accurate adjustment. It is claimed that operating both magnetos tends to increase the power at high speed by igniting the charge at two points and thus accelerating ignition.

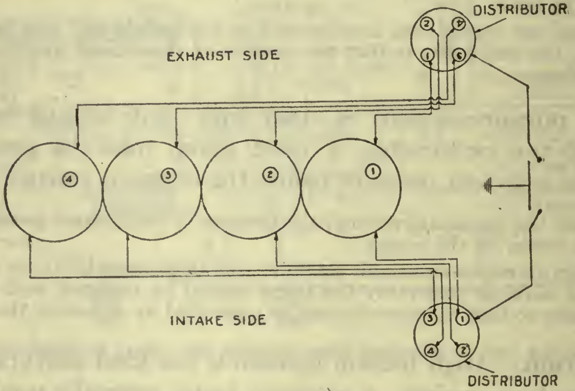


FIG. 3,080.—Wiring diagram for Dixie magneto ignition system as used on Hall-Scott four cylinder engine.

Because of the severe igniting conditions, spark plugs are particularly liable to fail, and in this connection it may be well to caution the operator not to let the spark gap get too large, maintaining the proper gap by frequent adjustment of the spark points.

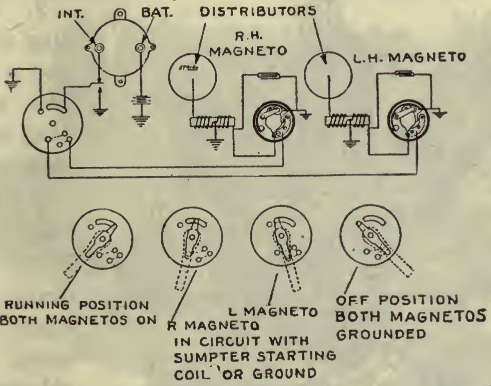
It should be remembered that the pressure necessary to force a spark across the gap increases with the length of the gap and the compression.

Because of the heavy compression, an unusually high voltage is required for the reason above, and if in addition the gap be too large, the pressure necessary to produce a spark may be increased to such an extent as to damage the coil insulation.

There are no *good* spark plugs—the best are only "fair," and accordingly the best that can be obtained hardly answer the purpose.

It should be understood that there is no reliable method of testing a spark plug without running the engine at full load so that the test may be made with maximum compression.

The practice of testing plugs by placing a screw driver in contact with the plug terminal and sparking across to the metal of the engine (the engine idling) is no test at all, except to prove that the plug will not spark under any conditions. This test is only of value when made with the engine running with full load (maximum compression) and this cannot be conveniently done either in an aeroplane or automobile.


INT.
BAT.
DISTRIBUTORS
R.H. MAGNETO
L.H. MAGNETO
RUNNING POSITION BOTH MAGNETOS ON
L MAGNETO
R MAGNETO IN CIRCUIT WITH SUMPTER STARTING COIL OR GROUND
OFF POSITION BOTH MAGNETOS GROUNDED

FIGS. 3,081 to 3,085.—Dixie magneto switch diagrams, and magneto connection as furnished for Hall-Scott four cylinder 100 horse power engine. The switch should be installed in the operator's seat. One lever controls the right hand and left hand magnetos. By shunting either one or the other, it can be quickly determined if both magnetos, with their respective spark plugs, be working correctly.

The reason the test is of no value is because, with little or no compression a spark will jump a gap of moderate length at about 5,000 volts, but if the throttle be opened wide with engine heavily loaded, it will require, because of the high compression, about 30,000 volts to produce a spark.

It can be readily understood that if the insulation be defective but adequate for 5,000 volts, it would allow current to pass for higher pressures, thus no spark would jump the gap between the spark points.*

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*NOTE.—A full treatment of both Carburetter and Ignition is given in the chapter on Gas ngines (pages 1,382 and 1,418 respectively) and a thorough study of these subjects is ecommended.

Care and Operation

To Insure Best Operation.—The following instructions should be carried out after every long flight. If used for schooling purposes, daily care should be taken, as follows:

1. While engine is still warm remove all spark plugs.

2. Clean each plug with gasoline and a stiff brush, and space each plug with proper .018 gauge.

3. Remove lower crank case sump plug, and drain oil into a clean measure, cover same and allow to stand until morning.

4. Clean out lower crank case thoroughly with kerosene.

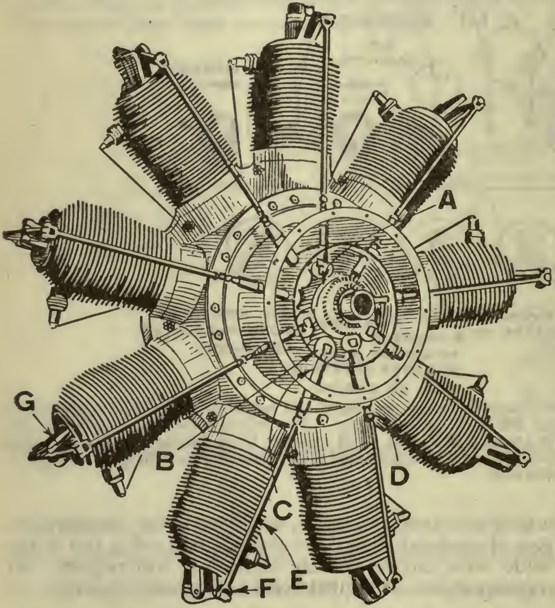


FIG. 3,086.—Cam action of Gnome engine. The nine holes in the crank pin feed oil to nine cams, one being shown at A. The hole is at the base on the incline in each case. ***In operation*** cam roller B, picks this oil up and carries it over the cam surface, some of it reaching the small oil holes on each side of these rollers and oiling the bearings of the rollers. The surplus oil from here feeds up through the valve rod guides, thence it feeds through ball joint D, through hollow valve rod E, and oils the pin at F. There is also sufficient oil at this point to strike a grove on the under side of the valve lever and feed along to the lever bearing G, so that every bearing is well oiled from the central supply. This, however, requires a large amount of oil, a characteristic of the rotating type of engine.

5. Squirt gun full of kerosene into each cylinder through spark plug opening.

6. Remove front portion magneto distributer cover and wipe out distributer block with soft cloth moistened in alcohol or gasoline if necessary.

7. In replacing distributer block, be sure to wipe off all excess oil on magnetos and cover before replacing.

8. Cover engine for night with canvas or heavy cloth.

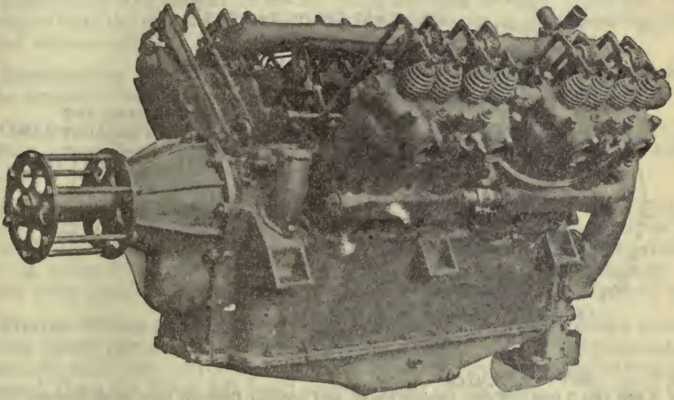


FIG. 3,087.—Front side of Sturtevant eight cylinder water cooled V engine. Cylinder 4×5½; *r.p.m.*, 2,000; horse power, 140. The propeller shaft is driven through reducing gears, the standard ratio being 5:3, giving a propeller speed of 1,200 *r.p.m.* The direct drive is especially adapted to dirigible work.

Preparing Engine for Service.—Give the engine a few turns and note that all working parts are perfectly free. Replace lower oil sump plug.

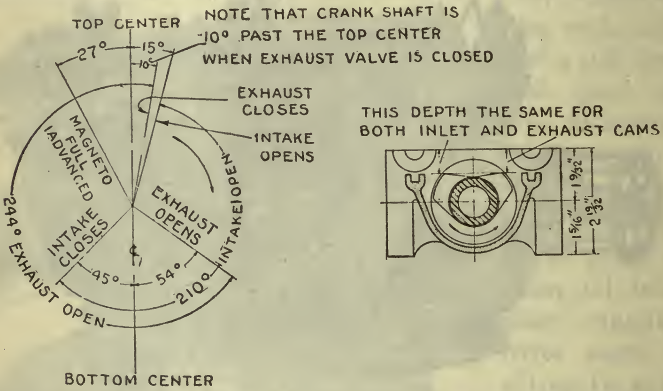
Pour off top oil, left to stand over night, into a clean measure, making sure that the heavier portion of oil and carbon deposit is left in first measure. Add to the second measure enough new oil to make two gallons.

Pour same into sump through breather pipe lead.
Replace spark plugs and connect lead wires.

The engine is now ready to start. Run slowly with engine throttled for at least five or six minutes while the plane is on the ground, before starting on flight, so that the lubricating oil will have a chance to work up onto the cylinder walls and pistons.

For normal flight do not place more than the amount of oil specified by the manufacturer in sump.

Standard Adjustments.—Of the various adjustments that must be made from time to time for satisfactory working the following should be noted.



FIGS. 3,088 and 3,089.—Valve timing chart, and section through cam shaft housing of Hall-Scott four cylinder 100 horse power engine, showing position of cams when exhaust valve is closed.

Spark Plugs.—These should have .018 inch clearance between points across which the spark jumps.

Magneto Breaker Points.—The gap between breaker points in magnetos when full open should be about .02. Use gauge furnished with each magneto screw driver. It might be possible to obtain better results if the breaker gap be closed to .018.

Oil Pressure.—This will vary according to weather conditions and gravity of oil used. In normal weather, with engine properly warmed up, the pressure will register upon the oil gauge from 5 to 15 lbs. when engine is turning from 1,275 to 1,300 *r.p.m.*

Air Pressure.—Air pressure gauge should not register under 3 lbs.

Valve Clearances.—Inlet and exhaust valves should be set with a .02 inch clearance, when motor is cold. This should be checked by a timing disc.

Gear Clearances.—With the exception of the lower pinion shaft gear meshing with the crank shaft, which has a .01 clearance; all other gears have a .02 clearance.

Bearing Clearances.—Crank shaft bearings should have a .001 clearance.

Connecting rods should be set up snug, allowing enough clearance, however, so that the rod may be slid laterally on the crank shaft bearing when oiled without binding.

Lubricating System.—(Diagram of lubrication system on page 1,606.) The proper lubrication of an airplane engine is of vital importance.

Oils best adapted for Hall-Scott engines have the following properties: a flash test of not less than 400 F.; viscosity of not less than 75 to 85 taken at 212 F. with Saybolt's universal viscosimeter.

Engine Oiling System.—Crank shaft, connecting rods and all other parts within the crank case and cylinder are lubricated directly or indirectly by a force feed oiling system. The cylinder walls are lubricated by oil spray thrown from the lower end of connecting rod bearings. A small tube supplies oil from connecting rod bearing directly upon wrist pin.

Engine Oiling Circulation.—The oil is drawn from the strainer located at the lowest portion of the lower crank case, forced around the main intake manifold oil jacket. From here it is circulated to the main distributing pipe located along the lower left hand side of upper crank case. The oil is then forced directly to the lower side of crank shaft through holes drilled in each main bearing cap. Leakage from these main bearings is caught in scuppers placed upon the cheeks of the crank shafts furnishing oil to the connecting rod bearings. Small tubes leading from the bearings convey the oil to the wrist pins.

Draining Oil from Crank Case.—The oil strainer plug is located at the lowest point of the lower crank case. This is a combination dirt, water, and sediment trap. It is easily removed by unscrewing. This strainer should be removed after every five to eight hours running of the engine and cleaned thoroughly with gasoline. It is also advisable to squirt distillate up into the case through the opening where the strainer has been removed.

Allow this distillate to drain out thoroughly before replacing the plug with strainer attached. Be sure gasket is in place on plug before replacing.

Pour new oil in through the breather pipe on exhaust side of engine. Be sure to replace breather pipe strainer screen, if removed.

Insufficient Lubrication.—If, through oversight, the engine do not receive sufficient lubrication and begin to heat or pound, it should be stopped immediately. After allowing engine to cool, pour at least two gallons of oil into oil sump. Fill radiator with water after engine has cooled. Should there be apparent damage, the engine should be thoroughly inspected immediately, without further running.

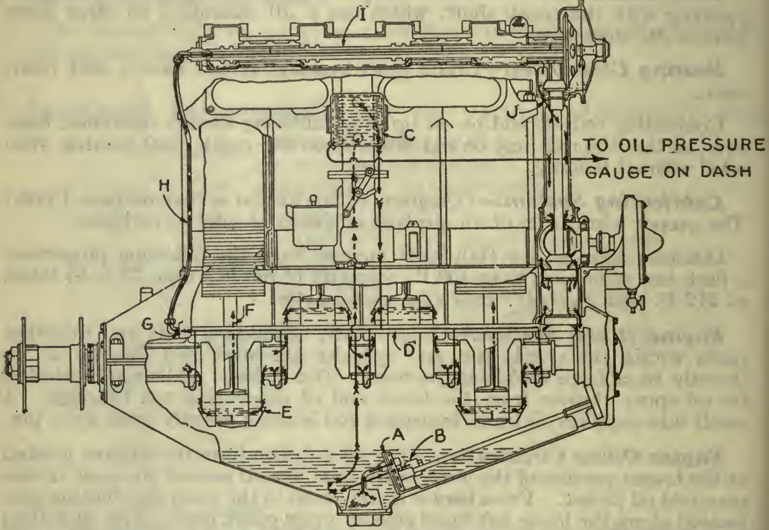
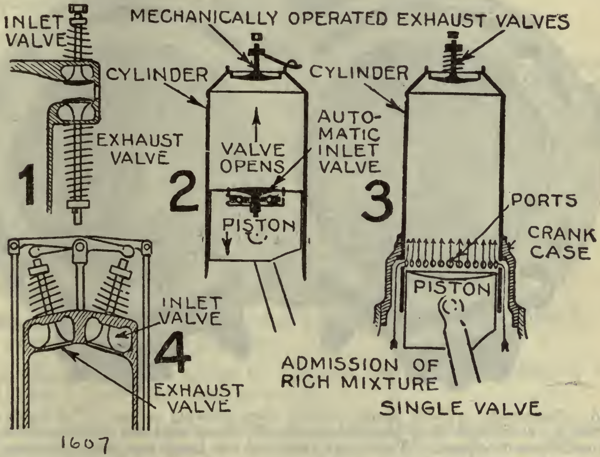


FIG. 3,090.—Oil flow diagram of Hall-Scott 100 horse power four cylinder engine. ***In operation,*** gear pump at A, pumps the oil through pressure relief valve at B, into oil jacket surrounding intake manifold at C. From the manifold oil jacket, the oil goes to the oil main in upper crank case at D. From the main individual leads, the oil is forced to each main bearing delivered through the bearing caps to the under side of the shaft. The oil that escapes at the end of the main bearings is thrown outward along the crank cheeks, being caught by the scuppers at E, and directed into the hollow crank pin. A hole in the crank pin wall allows the oil to be forced by centrifugal pressure onto the connecting rod bearing. A tube located on the side of the connecting rod at F, conducts the oil from the crank pin bearing onto the wrist pin bearing. Cylinder walls and pistons are oiled by splash from the crank shaft and connecting rods. At G, in the end of the oil main, is located an orifice which allows a measured flow into the pipe H, which carries the oil to one end of the cam shaft housing and forces it into the bore of the hollow cam shaft I. Holes through the cam shaft wall allow the oil to flow onto the bearings. Oil escaping from the cam shaft bearings gathers in the bottom of the cam shaft housing and flows to the forward end where it forms a bath for the cam shaft gears and upper vertical shaft bearings. A by-pass at J, carries the oil into the vertical shaft housing. Descending through the vertical shaft housings, the oil thoroughly lubricates all gears and ball bearings used in driving the magnetos, water pump and pinion shaft. The oil pressure on the bearings is registered by a dash gauge. Care should be taken that the oil does not work up into the gauge as it will prevent the correct registering of the pressure.

If no obvious damage has been done, the engine should be given a careful examination at the earliest opportunity to see that the running without oil has not burned the bearings or caused other trouble.

Auxiliary Oil Tank.—For long flights, an auxiliary oil tank is useful. This tank should be placed slightly higher than the lowest point of the engine oil sump, thus allowing new oil, if required, to flow from this tank into the main sump, so that the oil will flow by gravity through the lead pipe to main oil sump on engine.

The crank case oil sump has a capacity sufficient for from three to six hours' flight, depending entirely upon weather conditions, kind of oil used and *r.p.m.* of engine.



FIGS. 3,091 and 3,094.—Arrangement of inlet and exhaust valves on aeroplane engines. 1, super-imposed; 2, Gnome inlet in piston and overhead exhaust; 3, so-called single valve; 4., overhead inlet and exhaust.

Preparation to Start Engine.—Always replenish gasoline tanks through a strainer which is clean. This strainer must catch all water and other impurities in the gasoline.

Pour the proper amount of fresh oil (as recommended by the manufacturer of the engine) into the lower crank case.

Oil all rocker arms.

Be sure radiators are filled within one inch of the top.

After all the parts are oiled, and the tanks filled, the following must be looked after before starting:

See if crank shaft flange be tight on shaft.

See if propeller be trued up to within ⅜ inch.

See if propeller bolts be tight and evenly drawn up.

See if propeller bolts be wired.

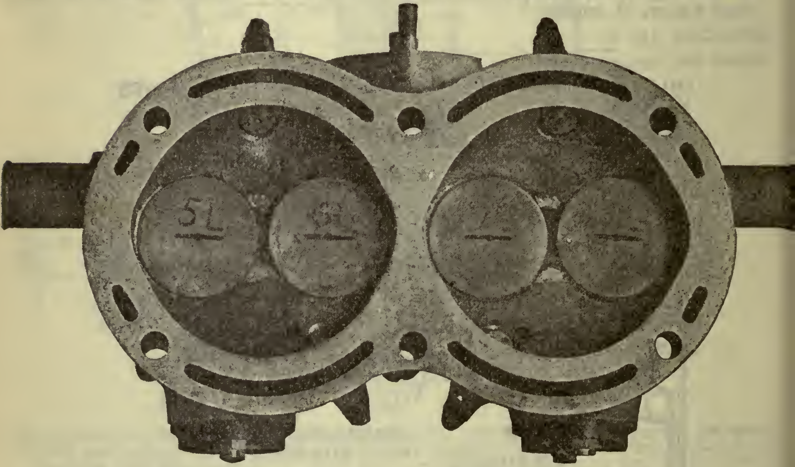


FIG. 3,095.—Sturtevant cylinder heads; interior view showing valves. The heads are cast in pairs from an aluminum alloy and contain ample water passages for circulation of cooling water over the entire head. The water jacket of the heads has large openings in both to allow the unobstructed circulation of the cooling water. The cylinder heads and cylinders are both held to the base by six long bolts.

Oil Pressure Relief Valve.—An oil relief valve is located just off of the main oil pump in lower crank case. This regulates the pressure at all times, so that in cold weather there will be no danger of bursting oil pipes due to excessive pressure. If it be found that the oil pressure is not maintained at a high enough level, inspect this valve. A stronger spring will not allow the oil to by-pass so freely and consequently the pressure will be raised; a weaker spring will by-pass more oil and reduce the oil pressure materially.

Every four days the magnetos should be given two or three drops of

Every month all cylinder hold down nuts should be gone over to ascertain if they be tight. (Be sure to re-cotter nuts.)

See if magnetos be bolted on tight and wired.

See if magneto cables be in good condition.

See if rocker arm tappets have a .02 clearance from valve stem when valve is seated.

See if tappet clamp screws be tight and cottered.

See if gasoline, oil, water pipes and connections be in perfect condition.

Air on gas line should be tested for leaks.

Pump at least three pounds air pressure into gasoline tank.

After making sure that above rules have been observed, test compression of cylinders by turning propeller.

"*Do not forget to short both magnetos.*"

Be sure compression relief cocks do not leak. If they do, replace same with new ones immediately, as compression leak might cause premature firing.

Open priming cocks, located upon the intake manifold, and squirt some gasoline into each.

Close cocks.

Open throttle slightly.

Retard magnetos.

Open magneto switch.

Starting.—If all the foregoing directions have been carefully followed, the engine is ready for starting.

In cranking engine either by starting crank, or propeller, it is essential to throw it over compression quickly.

Care should be taken when mounting the propeller to locate it in a suitable position so that it will be possible to crank it rapidly over compression.

If the engine be started with compression release cocks open, they should be closed immediately after starting.

When engine is running, advance magnetos.

Do not accelerate engine until it has been warmed up thoroughly.

Allow the engine to warm up gradually.

After this has been done short one magneto and then the other, to be sure both magnetos and spark plugs are firing properly. If there be a miss, the fouled plug must be located and cleaned.

The oil gauge provided is an exceedingly delicate instrument which might easily be broken by excessive pressure caused by rapid acceleration of a cold engine. Care should be taken to see that oil does not get up inside of gauge which will stop it up, usually making the hand drop to zero.

In cold weather the radiator should be covered until the temperature of the water reaches 80 to 100 degrees Fahr.

To Stop Engine.—Before stopping engine, let it run slowly for a minute or two, then short magnetos. It is advisable after a long run, while the engine is still hot, to squirt kerosene into the exhaust valve guides. Five or ten minutes after this is done, the engine should be re-started and run three or four minutes before stopping. This will insure an easy start next morning. It will also loosen up the carbon deposit upon the pistons and exhaust valve stems.

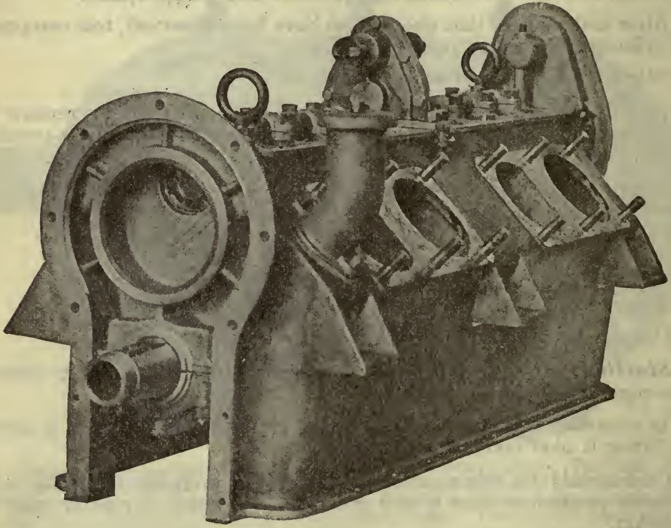


FIG. 3,096.—Sturtevant crank case. It is cast from aluminum alloy. The sides extend considerably below the center line of the crank shaft giving a deep section.

Irregularities in Starting Engine.—Failure to start the engine might be caused by the throttle being open too wide. In cold weather the carburetter should be primed well; and in warm weather should be primed slightly. After the engine has once started, it should not be primed too much before re-starting. The use of high test gasoline for priming helps starting greatly, but should be used with caution, taking care not to over prime cylinders, allowing gas to run down into crank case, which might cause considerable damage.

After repeated attempts to start engine as just directed, and not succeeding, it is advisable to inspect the spark plugs. Five out of ten cases of ignition trouble are due to dirty or incorrectly spaced spark plugs. The correct spacing is .018. All should be uniform. Be sure points are clean. Dip tooth brush in gasoline and brush points. Examine the wire leads to be sure they are each in their proper place.

If, after a test, it be found that there is no current delivered at the spark plugs, the magneto should be looked into according to the instruction book issued by the magneto manufacturer, and if not easily remedied they should be sent to the factory representative for repairs.

If the ignition system be found satisfactory, the trouble must then be in the fuel supply, or the fuel itself. The carburetter float may be full of gasoline, allowing the float chamber to flood, which will also happen if the needle valve do not work free, or some obstruction get in between seat and valve. The supply pipe might be clogged partially or altogether. If force feed be used, the air pressure must be tested, as there might be an air leak, or the air pressure gauge might register a certain amount of air which is not in the tank. There is a possibility that the jets in the carburetter are stopped up. If this be the case, do not attempt to clean same with any sharp instrument. If this be done, it might change the opening in the jets, thus spoiling the adjustment. Jets and nozzles should be blown out with air or steam.

FIGS. 3,097 to 3,099.—Sturtevant connecting rod piston, piston rings, and wrist pin

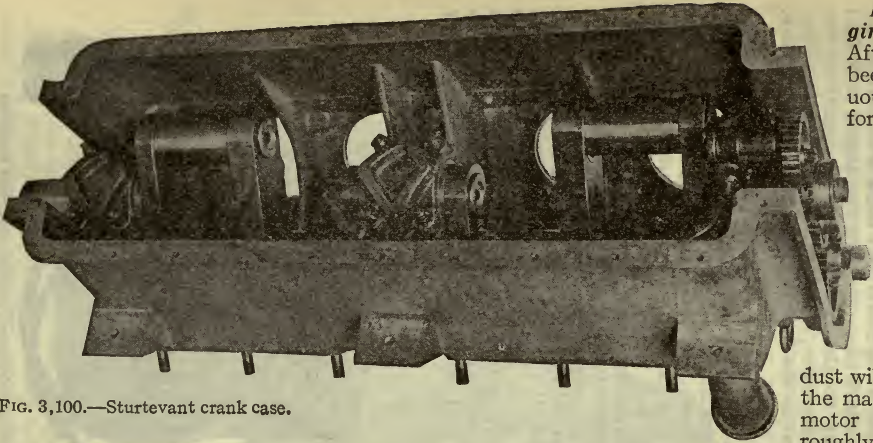


FIG. 3,100.—Sturtevant crank case.

Decrease in Engine Efficiency.—After the engine has been used continuously in hard service for a certain length of time, it may decrease in its *r.p.m.* There are different causes for this. In the magneto distributer, the carbon brush running between the distributing points will leave a deposit. The fine dust will partially shorten the magneto, causing the motor to prefire or run roughly.

The distributer block should be removed and wiped out as previously suggested.

Carbon in the cylinders and top of the pistons will also decrease the *r.p.m.*, as well as cause pre-ignition, hammering and loss of compression. If proper engine oils be used this trouble will be delayed for the longest possible time. Natural wearing of surfaces must be taken into consideration, and accordingly, the disassembling, inspection and cleaning of engine within a reasonable length of time.

It is advisable to remove the propeller if undue vibration from the engine be experienced and check up as to its pitch, balance, etc., as an engine will run roughly if the blade be out of balance or not true to pitch.

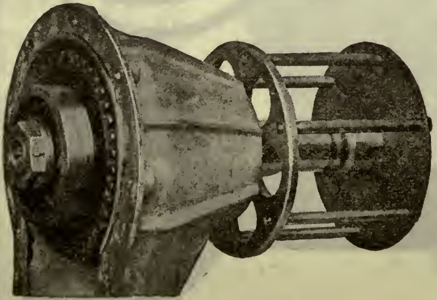


FIG. 3,101.—Sturtevant propeller shaft.

CHAPTER 51

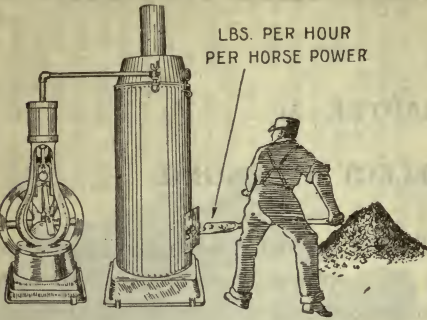
HOW TO SELECT AN ENGINE

It usually happens that the one entrusted with the important task of choosing an engine for a power installation, has very little knowledge of engines in general and is therefore unqualified to make an intelligent selection. Moreover, those furnishing the money, as a rule have little or no knowledge of machinery, and consequently do not appreciate the great loss entailed by installing the wrong type, make, or size of engine.

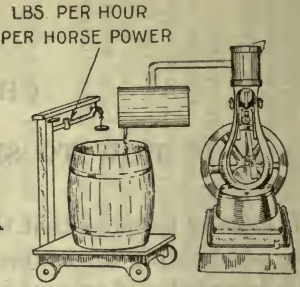
In order to properly select an engine, one should have a considerable knowledge of:

1. Steam engine economy;
    - *a*. Factors governing economy.
    - *b*. Feed water consumption.
    - *c*. Quality and pressure of steam required.
2. Steam engine construction;
    - *a*. Service requirements.
    - *b*. Durability { proportion of parts
    - *c*. Relative cost of different types.
3. Type of boiler and auxiliary machinery involved:
4. Plant conditions;
    - *a*. Nature of the load.
    - *b*. Growth of plant.
    - *c*. Uses for steam.
    - *d*. Quality and cost of the water, fuel.
    - *e*. Quality and cost of the fuel.
    - *f*. Miscellaneous supplies.
    - *g*. Space required.
    - *h*. Attendance.
    - *i*. Interest on investment.
    - *j*. Depreciation.

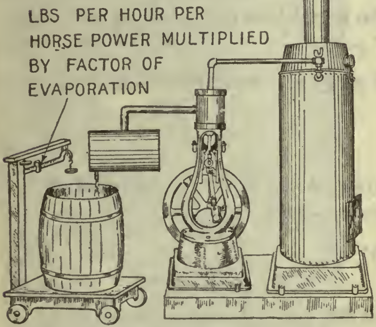
It must be evident to anyone that the statement that one engine requires only ½, ⅓ or even ⅕ as much fuel as some other engine means nothing unless the following items for both engines be known:



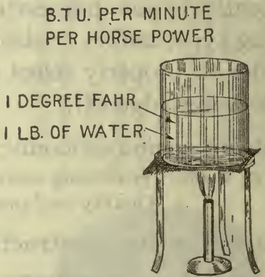
Coal Unit



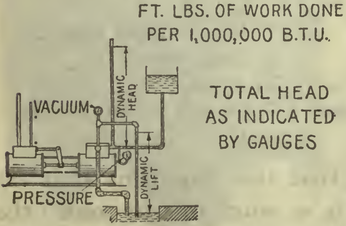
Steam Unit



Equivalent Steam Unit



Heat Unit



Duty Unit

Figs. 3,102 to 3,106.—Various units of engine performance graphically defined. Fig. 3,102. ***coal unit;*** fig. 3,103, ***steam unit;*** fig. 3,104, ***equivalent steam unit;*** fig. 3,105, ***heat unit;*** fig. 3,106, ***duty unit.*** In fig. 3,102, the fireman shovels about 6 lbs. of coal per hour per horse power, and assuming the engine to be say 10 horse power, would make a total of 60 lbs. of coal per hour required. Fig. 3,103 shows the method of measuring feed water of a condensing engine, and fig. 3,104, the same reduced to a "*from and at* 212° F." basis. Similarly, figs. 3,105 and 3,106, point out things to be considered in the *heat* and *duty* units of engine performance.

1. Cost of fuel and water.
2. Cost of plant.
3. Attendance required.
4. Depreciation.

Thus, the owner of a saw mill may put in a multi-expansion condensing engine using high pressure superheated steam, involving expensive boiler, auxiliaries, skilled attention, etc., and, although he obtain a horse power hour on only 10 pounds of steam, the net expense may be more than if he installed the simplest and most wasteful type of engine, requiring 50 or 60 pounds of steam per horse power hour.

Units of Engine Performance.—There are several ways in which the *input*, required by an engine may be expressed, such as:

1. Pounds of coal per *hour* per horse power (***coal unit***).
2. Pounds of steam per hour per horse power (***steam unit***).
3. Equivalent pounds of steam per hour per horse power (***equivalent steam unit***).
4. British thermal units per minute per horse power (***heat unit***).
5. Fort pounds per 1,000,000 *B.t.u.* (***duty unit***).

Whereas the coal unit (pounds of coal per horse power hour) is what the owner of the plant is interested in, this is misleading and not fair to the engine, because the latter uses steam and not coal.

By knowing both the coal unit and steam unit, the performance of the boiler and engine are expressed independently, hence any undue waste can be located. Thus, a certain engine may require an abnormal amount of coal; by testing the engine it can be determined if the steam consumption be in proportion to the coal used, the result indicating either:

1. Losses in the engine by leakage or faulty steam distribution, or
2. Losses in the boiler due to scale, dirty tubes, improper firing, etc.

An important point to note in regard to the steam unit is that only a very poor idea of the actual economy of an engine can be obtained by simply stating the number of pounds of steam* it consumes per horse power hour without giving the *pressure* and *quality* of the steam, and temperature of the feed water.

---

*NOTE.—The reader should not confuse the term ***"pounds of steam," 1,*** when used as a measure of *quantity*, meaning *the **weight** of the water from which the steam was generated*, and ***2,*** when used as a measure of *quality*, meaning *its **pressure** per square inch*. Thus, a certain engine requires 32 pounds of steam (*weight*) per horse power hour with steam gauge registering 80 pounds of steam (*pressure*). See fig. 4,007.

Ques. Why is this?

Ans. Because simply the weight of steam generated does not indicate the amount of heat energy, and consequently the amount of coal required to generate the steam.

Ques. How are the relative demands on the boiler ascertained?

Ans. By finding the *relative steam unit*.

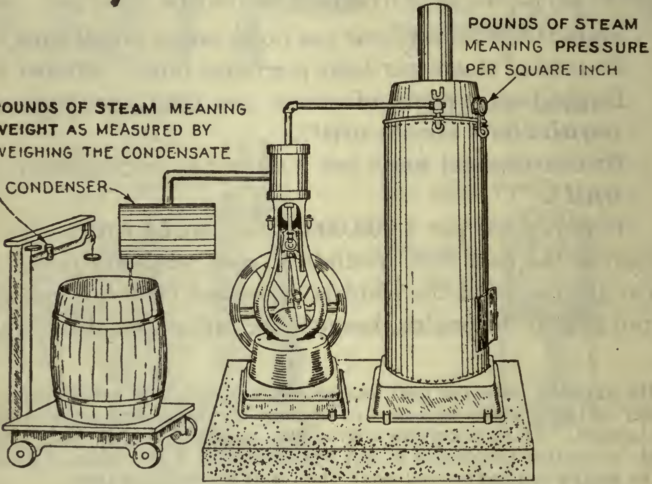
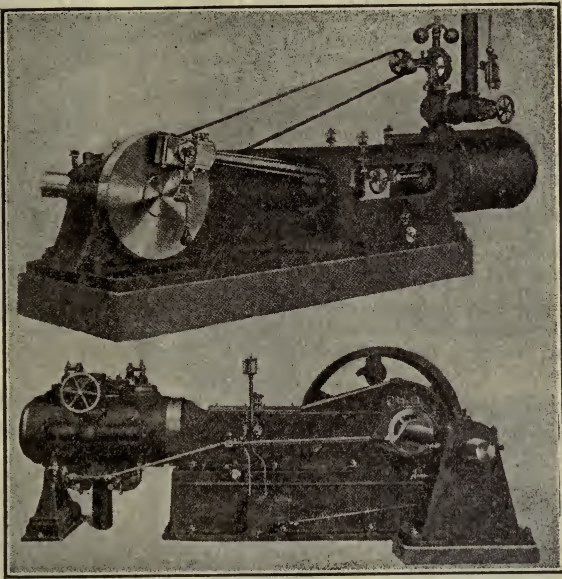

POUNDS OF STEAM MEANING PRESSURE PER SQUARE INCH
POUNDS OF STEAM MEANING WEIGHT AS MEASURED BY WEIGHING THE CONDENSATE
CONDENSER

FIG. 3,107.—Distinction between the terms "*pounds of steam*" meaning ***pressure,*** and "*pounds of steam*" meaning ***weight.*** Thus, the type of engine and boiler shown would ordinarily be run with 80 *pounds of steam* (***boiler pressure*** per square inch) and would require about 50 *pounds of steam* (***weight***) per horse power per hour. This quantity is usually determined in testing by passing the exhaust through a condenser and weighing the condensate in a barrel on a platform scale, deducting of course the weight of the barrel to obtain the net weight of the water. The pounds of condensate discharged into the barrel per hour divided by the horse power gives the pounds of steam used by the engine per horse power per hour and is a measure of its economy.

That is to say, the equivalent evaporation *from and at 212° Fahr.** corresponding to the steam units as illustrated in the example following:

---

*NOTE.—The term ***from and at 212° Fahr.***, means that the steam is generated at a temperature of 212° from the water of the same temperature. Steam is generated at 212°, if the vessel containing the water be open to the atmosphere and the barometer read 29.921, the pressure corresponding to this reading being known as the *standard atmosphere*.



FIGS. 3,108 and 3,109—Two engines ***illustrating*** the term ***pounds of steam*** in its economic sense. The throttling engine fig. 3,108 is very wasteful and will require about 50 *pounds of steam* per horse power per hour, while the uniflow engine shown in fig. 3,109, being designed for economy will run on only about 15 *pounds of steam* per horse power per hour.

NOTE.—The following table for Fernald and Orrok serves to illustrate the term "*pounds of steam*," and also the marked gain in steam consumption of condensing engines over non-condensing. The figures given are approximate only. The student of steam engine economy should make a study of the tabulated results of a large number of tests, noting in each case all the particulars of the engine.

Pounds of Steam

(Per indicated horse power per hour)

TYPE OF ENGINE	Non-Condensing		Condensing		Per cent Gained by Condensing
	Probable Limits	Assumed for Compression	Probable Limits	Assumed for Expansion	
Simple high speed	65–25	33	50–19	23	30
" low speed	30–22	25	24–17	19	24
Compound high speed	30–22	26	24–16	20	23
" low speed	24–18	21	20–12	17	19
Triple high speed	27–17	22	23–14	17	23
" low speed			18–11	16	

Example.—Two engines, ***A*** and ***B,*** require steam as follows: ***A,*** 15 pounds of steam at 200 pounds abs. pressure, 200° superheat; feed water 110°. ***B,*** 17 pounds of saturated steam at 100 pounds abs. pressure; feed water 200°. What are the relative demands on the boiler?

	Total heat above 32°	Feed water	Latent heat at 212°	Factor of evaporation
*Engine **A***	(1307.7	− 110)	÷ 970.4 =	1.234
*Engine **B***	(1186.3	− 200)	÷ 970.4 =	1.015

Equivalent evaporation from and at 212° Fahr.

*Engine **A.*** 15 × 1.234 = ... 18.5 pounds
*Engine **B.*** 17 × 1.015 = ... 17.3 pounds

Thus, it is seen that while engine ***A,*** requires 2 pounds of steam *less* than engine ***B,*** the actual equivalent evaporation required of the boiler is 18.5 − 17.3 = 1.2 pound *more* for ***A,*** than for ***B.***

Ques. In what other way is the boiler demand for various engines reduced to a common basis?

Ans. By measuring the steam consumption in *heat units per indicated horse power per minute.*

Example.—An engine consumes 20 pounds of steam when running condensing with saturated steam of 100 pounds abs. pressure; temperature of feed water 110° Fahr. How many *B.t.u. per minute* are being consumed by the engine?

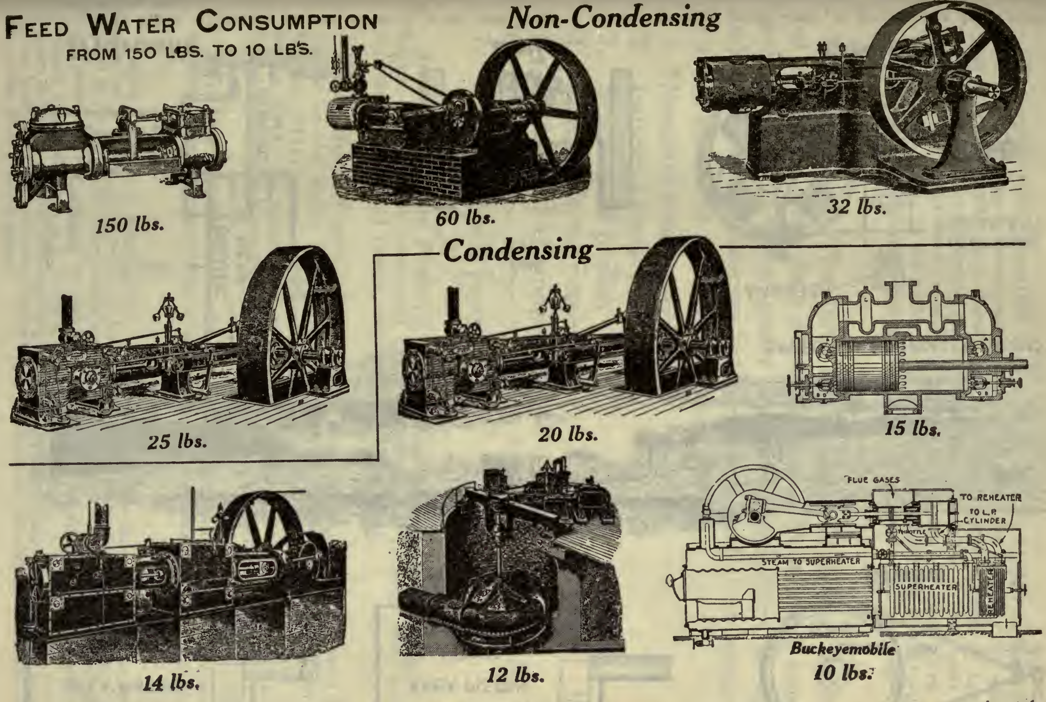
Total heat in one pound of steam at 100 pounds als.
(from table) = ... 1186.3 *B.t.u.*

Total heat above 32° Fahr. in one pound of the feed
water at 110° = 110° − 32° = ... 78 *B.t.u.*

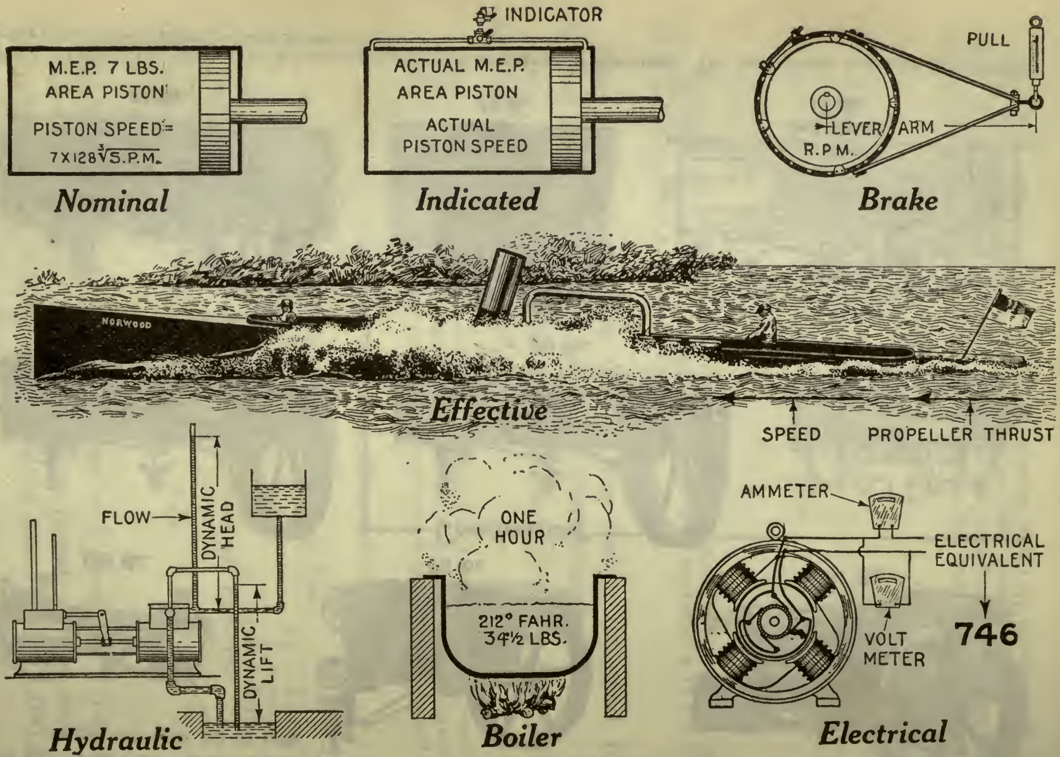
Heat supplied by boiler per pound of steam........ 1108.3 *B.t.u.*
Heat consumed by the engine per *horse power* ***per minute.***

$$\frac{1108.3\times 20}{60}=369.4\ B.t.u.$$

The term duty is used in connection with pumps as a measure of economy. Duty is defined as *the foot pounds of work done per 100 pounds of coal, or per 1,000 pounds of steam, or per 1,000,000 B.t.u. furnished by the boiler.*



FIGS. 3,110 to 3,118.—Various engines arranged according to ***degrees of economy.*** The figures below each give approximately the *lbs. of steam required per hour per horse power.*



FIGS. 3,119 to 3,125.—Principal factors involved in the calculation of various kinds of horse power, as described in detail in the accompanying text.

The power test committee of the A. S. M. E. (1915) defined duty as follows: *The duty per million heat units is found by dividing the number of foot pounds of work done during the trial by the total number of heat units consumed, and multiplying the quotient by 1,000,000.* This unit is the equivalent of 100 pounds of coal when each pound of coal gives 10,000 *B.t.u.* to the water in the boiler, or where the evaporation is $10{,}000 \div 970.4 = 10.305$ pounds of water from and at 212° Fahr. per pound of fuel.

Horse Power.—The term horse power is defined as *33,000 foot pounds of work done in one minute*. There are, however, several kinds of horse power and it is important to clearly understand the differences between these and their application to avoid confusion and errors. They are:

1. *Nominal horse power.
2. Indicated horse power.
3. Brake horse power.
4. Effective horse power.
5. Hydraulic horse power.
6. Boiler horse power.
7. Electrical horse power.

Indicated Horse Power.—This represents the power developed in the engine cylinder as measured by the indicator. It does not, however, represent the useful power delivered by the engine.

Brake Horse Power.—This is the *actual* horse power delivered by the engine to the shaft, being equal to the *indicated horse power* **minus** *the friction of the engine*. It is measured by applying a dynamometer or friction brake to the fly wheel (hence the name); though usually known as *brake* horse power it is sometimes called *delivered* horse power.

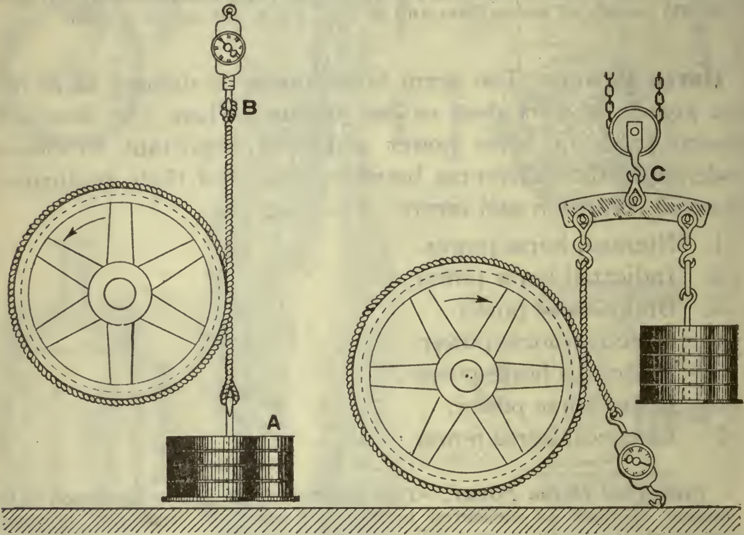
Effective Horse Power.—In any system of which the engine is only one element, the effective horse power is *the actual horse power given* ***by*** *or* ***to***

---

*NOTE.—***Nominal horse power.***—This term, which is due to Watt, has long been obselete but it should, however, be understood. Watt found that the mean pressure usually obtained in the cylinders of his engines was 7 pounds per square inch. He had also fixed the proper piston speed at $128 \times \sqrt[3]{\ }$ strokes from which he calculated the power which would be developed = area of piston $\times\ 7 \times 128 \times \sqrt[3]{\ }$ strokes per minute $\div$ 33,000. With subsequent increases in boiler pressures with resulting increases in mean pressures, the difference between nominal horse power and the actual horse power developed by the engine has become so great as to render Watt's rating useless.

the system. Thus, in a locomotive, the effective horse power given by the system (locomotive) to the cars is:

$$\frac{\textit{pull on tender draw link} \times \textit{speed}}{33{,}000}$$



FIGS. 3,126 and 3,127.—Friction rope brakes. (*Trans. A. S. M. E.*, vol. 24, p 739.) In the self-adjusting rope brake, fig. 3,126, if the friction at the rim of the wheel increase, it will lift the weight **A,** which action will diminish the tension in the end **B,** of the rope, and thus prevent a further increase in the friction. The same device can be used for a band brake of the ordinary construction. Where space below the wheel is limited, a cross bar **C,** supported by a chain tackle exactly at its central point, may be used as shown in fig. 3.127, thereby causing the action of the weight on the brake to be upward. A safety stop should be used with either form, to prevent the weights being accidentally raised more than a certain amount. To compute the horse power, multiply the difference between the weight **A,** and the weight shown on spring balance **B,** by the number of revolutions of the pulley per minute, and by the circumference of a circle passing through the center of gravity of the rope expressed in feet; finally dividing the product by 33,000. The radius is measured at the point where that part of the rope under highest tension leaves the rim of the wheel. If the self-adjusting feature be omitted, the weight may be determined by the use of platform scales, the two ends of the rope being attached to a vertical stand supported by the platform, and provision being made for adjusting the tension.

Again, in a steamship the effective horse power given by the engine to the system (power plant and vessel) is:

horse power delivered to shaft — loss due to slip.

Hydraulic Horse Power.—This may be defined as *the horse power required to elevate or force water against pressure;* it is as usually measured the indicated horse power at the water end of a pump. In calculations of hydraulic horse power required to elevate water, it is necessary to understand the difference between:

1. Static head,
2. Dynamic head,
3. Static lift,
4. Dynamic lift.

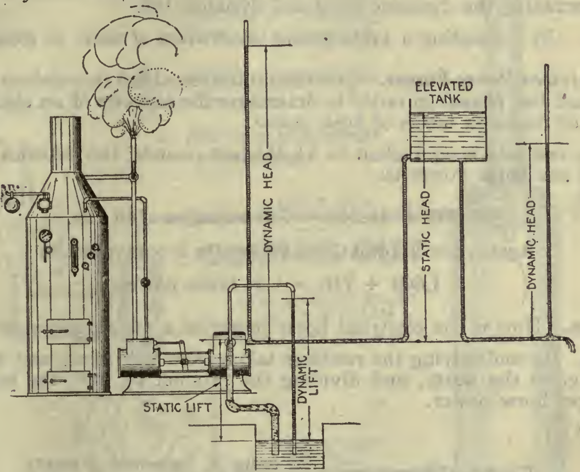


FIG. 3,128.—View of elevated tank with pump in operation maintaining the supply which is being drawn upon as shown, illustrating the terms static lift, dynamic lift, static head, and dynamic head. ***Static head** is the height from a given point of a column, or body of water **at rest**,* considered as causing or measuring pressure. ***Dynamic head** is an equivalent or virtual head of water in motion which represents the resultant pressure due to the height of the water from a given point, and the resistance to flow due to friction.* Thus, when water is made to flow through pipes or nozzles there is a loss of head. Similar definitions apply to ***static lift,*** and ***dynamic lift.*** In the figure the dynamic head is *greater* than the static head in the supply line to the tank, and *less* in the tank discharge line because of frictional resistance to the flow of the water. In ordinary calculation, it is common practice to estimate that every foot head is equal to one-half pound pressure per square inch, as this allows for ordinary friction in pipes.

Static head is the difference in elevation between the pump and height to which the water is raised.

Dynamic head is the *equivalent* head allowing for increase in pressure due to the friction of the water in flowing through the pipes.

Static lift is the difference in elevation between the pump and the level of the water supply.

Dynamic lift is the equivalent lift allowing for the increase (due to friction) in pressure necessary to force the water from the supply to the pump.

These various terms are illustrated in figs. 4,002 to 4,006, which should be thoroughly understood.

Ques. In calculating hydraulic horse power what should be considered as producing pressure?

Ans. The total dynamic lift or total equivalent distance the water is to be elevated being equal to *dynamic head* + *dynamic lift.*

Ques. How is the allowance which must be made for friction obtained in determining the dynamic head and dynamic lift.

Ans. By consulting a table giving the *friction of water in pipes.*

Electrical Horse Power.—It is desirable to establish the relation between *watts* and *foot pounds* in order to determine the *capacity* of an electric generator or motor in terms of *horse power.*

Since one *watt* is equivalent to 44.244 foot pounds, the electrical equivalent of one horse power is:

$$33{,}000 \div 44.244 = 746 \text{ watts, or } .746\ k.w.$$

Again, one kilowatt or 1,000 watts is equivalent to

$$1{,}000 \div 746 = 1.34 \text{ horse power.}$$

Ques. How is the electrical horse power of a motor obtained?

Ans. By multiplying the readings taken from volt meter and ammeter, which gives the watts, and dividing the product by 746, the number of watts per horse power.

That is:

$$\text{Electrical horse power} = \frac{\text{volts} \times \text{amperes}}{746} = \frac{\text{watts}}{746}$$

Efficiency.—In general the term efficiency may be defined as *the ratio of the useful work performed by a prime mover to the energy expended*, that is, *the* **output** *divided by the* **input.**

In any power plant, however, it is important to distinguish between several kinds of efficiency, as:

1. With respect to the engine;
   - *a.* Thermal efficiency.
   - *b.* Mechanical efficiency.
   - *c.* Combined efficiency.

2. With respect to the boiler;
   - *a*. Efficiency of the grate.
   - *b*. Efficiency of the furnace.
   - *c*. Efficiency of the heating surface.
   - *d*. Efficiency of the covering or insulating material.
   - *e*. Combined or overall efficiency covering items *a*, *b*, *c*, and *d*.
3. With respect to entire plant.
   - *a*. ***Net*** or ***commercial efficiency.***

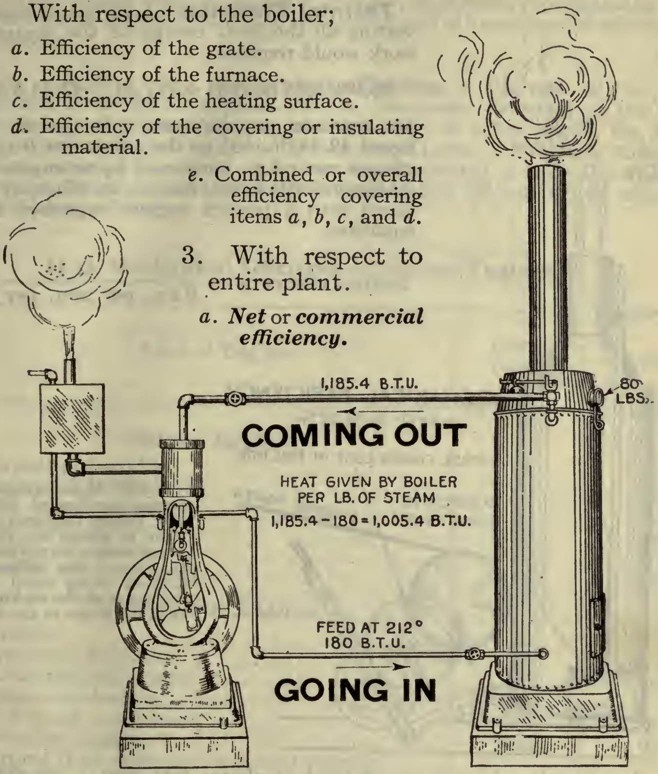

1,185.4 B.T.U.
80 LBS.
COMING OUT
HEAT GIVEN BY BOILER PER LB. OF STEAM
1,185.4 − 180 = 1,005.4 B.T.U.
FEED AT 212°
180 B.T.U.
GOING IN

FIG. 3,129.—***Thermal efficiency.*** The heat chargeable to an engine per lb. of steam is the difference between the total heat in a lb. of steam (***coming out***) at the boiler pressure and that in a lb. of the feed water ***going in*** the boiler. In the above case then, heat given by the boiler per lb. of steam $=1005.4$ *B.t.u.* Now the *B.t.u.* per ***minute*** used by the engine per lb. of steam per *i.h.p.* hour $=1005.4 \div 60 = 16.76$ and if the engine require 50 lbs. of steam per *i.h.p.* hour, then *B.t.u.* per *minute* per *i.h.p.* $=16.76 \times 50 = 838$, and the ***thermal efficiency***—$42.44 \div 838 = .05$, that is 5%.

Efficiency of the Engine.—The kind of engine efficiency that is most important is the combined efficiency or the ratio of useful work performed by the engine to the energy expended—*the* ***output*** *divided by the* ***input.*** Clearly this involves both the *thermal* and *mechanical* efficiency which respectively take into account the *steam* and *friction* losses.

Thermal Efficiency.—A *perfect* engine converting all the heat energy of the steam into work would require:

$$33{,}000 \text{ foot pounds} \div 777.54 = 42.44\ B.t.u.$$

per ***minute*** *per indicated horse power.* This figure 42.44 divided by the number of *B.t.u.* per minute per *i.h.p.*, consumed by an engine gives its thermal efficiency, that is, its efficiency compared with a perfect engine; expressed as an equation:

$$\text{thermal efficiency} = \frac{42.44}{B.t.u. \text{ per min. per } i.h.p.}$$

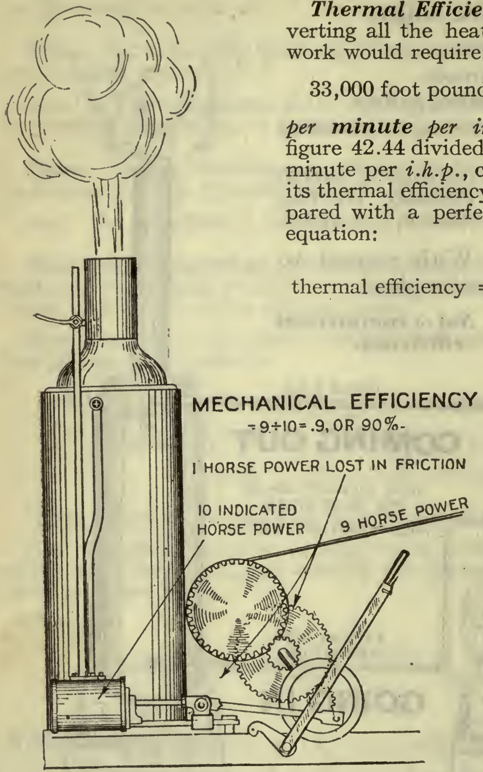


FIG. 3.130.—Hoisting engine illustrating mechanical efficiency. The power delivered at the drum is always *less* than the power applied to the piston by the steam; this difference, in amount being equal to the friction of all the working parts from the piston to the drum.

In practice *the thermal efficiency is always* **less** *than 1, the differenc representing the per cent losses, as by radiation, condensation, wire drawing, leakage, drop, etc.*

Mechanical Efficiency.—The statement that a given engine requires certain number of pounds of steam per indicated horse power per hour doe not represent the amount of energy expended in proportion to the usefu work done, that is, it does not take into account the losses due to frictio of the moving parts. Accordingly *the power delivered to the shaft wil always be* ***less*** *than the power applied to the piston*, the difference bein in amount equal to the power lost by friction. Thus, in a certain engin 5 per cent of the power is lost by friction, accordingly the power delivere to the shaft is 100% — 5% = 95% of the power supplied to the piston

This percentage, 95%, or .95, as it is usually expressed, represents the ratio of the power delivered to the power applied and is known as the mechanical efficiency.

By definition then, ***mechanical efficiency*** is the *ratio of the power delivered to the power applied*, that is, the power delivered to the shaft divided by the power applied to the piston—in other words, the brake horse power divided by the indicated horse power. Evidently then, the steam consumption of an engine as usually stated is only the *apparent* consumption and to obtain the true consumption the mechanical efficiency must be considered.

Mechanical Efficiency of Various Engines

Kind of Engine	Horse Power	Efficiency
Simple Engines		
Horizontal portable	25	.86
Horizontal portable	80	.91
High speed stationary	50	.92
High speed stationary	100	.94
Corliss, condensing	150	.85
Corliss, non-condensing	100	.86
Compound Engines		
Portable	80	.88
Horizontal stationary	75	.9
Horizontal mill engine	300	.86
Corliss, condensing	100	.9
High speed, condensing	46	.87
Pumping engines	650	.93
Vertical three cylinder compound electric lighting engines	6,000	.97
Triple Expansion Engines		
Vertical pumping engines	800	.94 to .97

Example.—A certain engine developing 100 indicated horse power, requires 25 pounds of steam per indicated horse power hour and its mechanical efficiency is .87. What is the real steam consumption?

The mechanical efficiency being .87, the power delivered to the shaft will be reduced to

$$100 \times .87 = 87 \text{ horse power}$$

but the total steam supplied to the engine per hour is

100 × 25 = 2,500 pounds

and as only 87 horse power is delivered to the shaft, the true steam consumption is

2,500 ÷ 87 = 28.7 pounds

instead of 25 pounds.

In general the mechanical efficiency of engines varies from .85 to .95.

The following table, by Fernald and Orrok, gives a few efficiencies obtained from tests of engines under normal but good working conditions:

Consumption of Condensing Engines

(Approximate steam consumption according to Kent)

*Type of engine	Steam Pressure							
	400	300	250	200	150	100	75	50
	Lbs. of steam per hour per horse power							
Ideal engine (Rankine cycle)	6.95	7.5	7.9	8.45	9.2	10.5	11.4	12.9
Quadruple expansion wastes 20 per cent	8.75	9.15	9.75	10.5	11.6	13.0	14.0	15.6
Triple expansion wastes 25 per cent	9.25	9.95	10.5	11.15	12.3	14.0	15.1	16.7
Compound wastes, 33 per cent	10.5	11.25	11.8	12.7	13.9	15.6	16.9	18.9
Simple engine wastes, 50 per cent	14.0	15.0	15.8	16.8	18.4	20.4	22.7	25.2

The following example will illustrate the importance of considering mechanical efficiency in selecting a boiler.

Example.—150 brake horse power is required to run a certain factory. If a compound condensing engine be installed having a mechanical efficiency of .86 and requires 21 pounds of steam per *indicated* horse power hour, initial pressure being 150 pounds and the feed water returned to boiler at 122° Fahr., what size boiler should be provided allowing an excess capacity of 25% for overload and steam supplied to auxiliaries, etc.?

Since 150 horse power must be delivered for the fly wheel and the mechanical efficiency is .86, the engine must develop

150 ÷ .86 = 174.4, say 175 *indicated* horse power

The total steam supplied to the engine is

175 × 21 = 3,675 pounds *per hour.*

Allowing 25% excess capacity, total steam to be supplied by the boiler is

3,675 × 1.25 = 4,594 pounds per hour *at* 150 *pounds pressure.*

---

*NOTE.—These engines are of the usual ratios of expansion. A 1 to 7 compound will be as economical as a 1 to 7 triple or quadruple expansion engine. It is conservative to say that compound engines may now be built to produce an indicated horse power on 12.5 lb. of saturated steam per hour. With high degree of superheat the 10-lb. mark has been passed but if results were calculated on the basis of saturated steam the figures would barely reach 10 lb. From 23 four-valve engines in commercial operation, Barrus reports: 1 falls below 12 lb.; 3 fall below 13 lb.; 16 fall below 14 lb.; only 3 fall above 14 lb.—*Fernald and Orrok.*

The factor of evaporation for steam at 150 pounds pressure generated from feed water having a temperature of 122° is (from table) 1.1388, hence equivalent amount of steam to be generated *"from and at 212°"* is

$$4{,}594 \times 1.1388 = 5.232 \text{ pounds per hour.}$$

The boiler capacity required then is:

$$5{,}232 \div 34\tfrac{1}{2} = 152 \text{ horse power.}$$

If in making the above calculation the mechanical efficiency were neglected, the boiler capacity would have been *less* than the above value in the proportion of:

$$1 - 21 \div (21 \div .86) = .14, \text{ that is } 14\% \text{ or } 20 \text{ horse power.}$$

That is, calculated boiler capacity would have been 152 − 21 = 131 horse power instead of 152 horse power.

Hence the energy output of the dynamo is

$$25 \times .83 = 20.8 \text{ horse power, or}$$

$$.746 \times 20.8 = 15.5 \text{ kw.}$$

The current output is

$$\frac{15.5 \times 1{,}000}{110} = 140.9 \text{ amperes.}$$

Boiler Efficiency.—In general, the efficiency of the boiler is defined as *the percentage of the total heat generated by the combustion of the fuel which is utilized in heating the water and generating the steam.* This is evidently the combined or overall efficiency, being the product of:

1. Efficiency of the grate.
2. Efficiency of the furnace.
3. Efficiency of the heating surface.
4. Effiicency of the boiler covering or insulating material.

The heat losses chargeable to the grate and furnace are those resulting from fuel dropping through the grates or withdrawn from the furnace, including the solid combustible matter in the cinders, sparks, flue dust, etc.

Considerable heat is lost due to bad proportion or condition of the heating surface.

The heat lost through the covering or insulating material of the boiler is not easily determined, especially if the boiler be enclosed in brickwork. The average loss for a brick setting is about 3%.

In determining the "efficiency" of a boiler (meaning the combined efficiency of the 4 above factors) the ratio is found between *the heat* *absorbed per pound of dry coal and the heat (calorific) value of one pound o dry coal*, or the ratio of *the two based on coal as fired.**

With hard coal the heating value of the combustible portion (the par that burns) is very nearly 14,800 *B.t.u.* per pound, equal to an evaporatio of $14{,}800 \div 970.4 = 15.25$ pounds of water *from and at 212° Fahr.*

A boiler which when tested with this coal shows an evaporation of 1 pounds of water per pound of combustible, has an efficiency of $12 \div 15.25 = 78.69\%$, as shown in fig. 3,131. This value is seldom approximated i practice.

The difference between the efficiency obtained by test and 100% is th sum of the various losses, the largest being the heat carried away by th chimney gases.

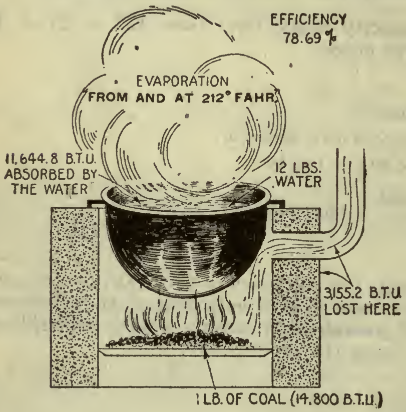


FIG. 3,131.—Evaporation "*from and a 212° Fahr.*", illustrating **boiler effi ciency** as explained in the text above Complete combustion is assumed.

Example.—The equivalent stean consumption (from and at 212°) of certain engine of 100 *i.h.p.* is 20 pound per horse power hour. Anthracite coa having a heating value of 14,800 *B.t.u* per pound is used. Assuming 15% passes through the grate as ash and unburned coal, how much coal is require per hour to run the engine?

Total steam required to run th engine is

$$20 \times 100 = 2{,}000 \text{ pounds per hour}$$

Since 15% of the coal passes through the grate the percentage available for combustion i

$$100\% - 15\% = 85\%$$

Since the heating value 14,800 *B.t.u.* is reduced to

$$14{,}800 \times .85 = 12{,}580 \text{ B.t.u. per pound of coal fired}$$

and the theoretical evaporation (from and at 212°) under these conditions is

$$12{,}580 \div 970.4 = 12.96 \text{ pounds per pound of coal fired,}$$

the amount of coal required to run the engine is

$$2{,}000 \div 12.96 = 15.43 \text{ pounds per hour}$$

---

*NOTE.—The weight of *combustible* is sometimes used instead of weight of coal. The term *combustible* means the portion of the fuel that is burned—the total weight of coal fired less that which passes through the grate. Accordingly efficiency based on combustible furnishes an approximate means for comparing the results of different tests, when the losses of unburned coal due to grates, cleaning, etc., are eliminated. In coal analysis combustible is defined as *the fixed carbon and volatile matter, the moisture and ash being excluded*. By some writers it is called *coal dry and free from ash*, and by others, *pure coal*.

Combined Efficiency.—In any system comprising several elements, as an engine and boiler, an engine connected to a dynamo, or to a pump, etc., the combined efficiency (sometimes called the overall efficiency) is the factor which includes the efficiency of each element. By definition, the ***combined efficiency*** of a system composed of several elements is *the product of the efficiencies of each element.* Thus in a system composed of two elements, engine and boiler, the combined efficiency is the product of the engine efficiency multiplied by the boiler efficiency.

Example.—In a certain pumping plant, as shown in fig. 3,132, there is a power pump connected by belt drive to an engine. If the efficiency of the various elements be: boiler 73%; engine: thermal 15%, mechanical 95%, belt drive 98%, pump 95%, what is the combined or overall efficiency?

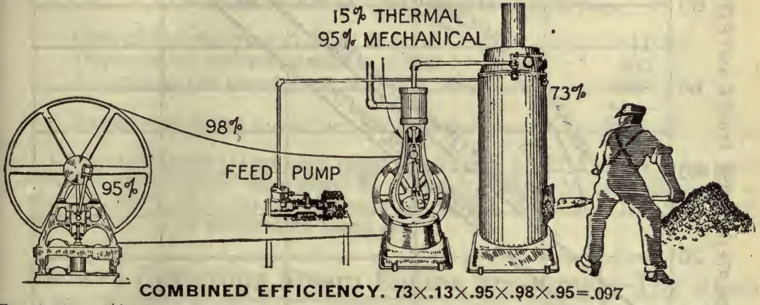


FIG. 3,132.—Diagram of simple steam plant illustrating combined efficiency. In the diagram 73% represents the efficiency of the boiler, including the pump.

The product of the above efficiencies is the combined efficiency, that is

$$\overset{\text{boiler}}{.73} \times \overbrace{.15 \times 95}^{\text{engine}} \times \overset{\text{belt}}{.98} \times \overset{\text{pump}}{.95} = .097 \text{ or } 9.7\%$$

Example.—In a 25 horse power 110 volt direct connected generator set the mechanical efficiency of the engine is .92 and efficiency of dynamo .9. What is the energy and current output of the dynamo?

As usually rated the value for power refers to the engine and here means 25 indicated horse power.

The combined efficiency then of the two elements is

$$.92 \times .9 = .83$$

Hence the energy output of the dynamo is

$$25 \times .83 = 20.8 \text{ horse power} \quad \text{or}$$
$$.746 \times 20.8 = 15.5 \text{ kw.}$$

The current output is

$$15.5 \times 1{,}000 \div 110 = 140.9 \text{ amperes.}$$

Net or Commercial Efficiency.—If the entire plant be regarded not merely from the standpoint of *combined efficiency* of the power system but also takes into consideration its *cost, depreciation, maintenance, etc.*, a true idea of the actual cost of the power is obtained. A factor may be found which takes into account all these items and which is called the *net efficiency*. It must be evident that this is a most important point, for upon it, depends *the **actual** cost of the power*.

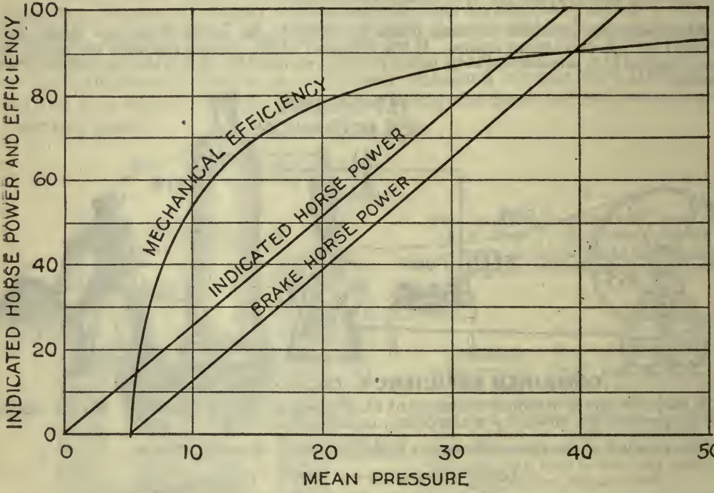


Fig. 3,133.—Mechanical efficiency curve. ***In construction,*** it is drawn for any engine by taking the value *brake horse power ÷ indicated horse power* for various mean effective pressures and setting off the value of the fraction to a vertical scale of percentage, and joining the points thus found. It will thus be evident that, so far as the mechanical efficiency is concerned, an engine should be worked up to its full load to obtain the maximum efficiency. The friction of an engine does, no doubt, to some extent increase with the load, but the proportional increase is so small as practically not to affect the result. The above remarks assume perfect efficiency of lubrication. Thurston gives the following values for the relative distribution of the friction in an engine with a balanced slide-valve: Main bearings, 47 per cent.; piston and rod, 32.9; crank pin 6.8; cross head and wrist pin, 5.4; valve and rod, 2.5; eccentric strap, 5.3 per cent. The frictional resistance of engines in general varies from about 8 per cent. to 20 per cent. of the full power.

Example.—Two types of power plants are to be considered for a certain proposed installation. ***Plant A:*** Cost $1,000; depreciation and maintenance, 5%; coal per horse power hour, 5 pounds. ***Plant B:*** Cost $5,000 depreciation and maintenance, 8%; coal, per horse power per hour, 1.5 pounds. With coal at $3.00 per ton, and interest at 6% on the investment, which plant is the more efficient *commercially* each operating 10 hours per day, 300 days per year?

Plant **A.** Coal used *per horse power* ***per year.***

$5 \times 10 \times 300 = 15{,}000$ pounds or 7.5 tons.

Cost of fuel per year = **7.5 × 5** =	37.5	
Depreciation and maintenance, 1,000 × .05	50	
Interest on the investment, 1,000 × .06	60	
Actual cost of the power per horse power year	$147.5	(1)

Plant **B.** Coal used *per horse power* ***per year.***

$1.5 \times 10 \times 300 = 4{,}500$ pounds or 2.25 tons.

Cost of fuel per year, 2.25 × 5 =	11.25	
Depreciation and maintenance, 5,000 × .08 =	400	
Interest on the investment, 5,000 × .06	300	
Actual cost of the power per horse power year	$711.25	(2)

Comparing items (1) and (2), plant **A** is $711.25 \div 147.5 = 4.8$ times more efficient *commercially* than plant **B.** That is, calling the net efficiency of plant **A,** 100%, then the net efficiency of plant **B,** is $147.5 \div 711.25 = .207$, that is, 20.7% as efficient as plant **A.**

Factors Governing Steam Engine Economy.—The steam and coal economy of engines is an important point in determining the relative value of different types and combinations of engines and boilers.

In general, cost depreciation, attendance, repairs, etc., increase with the economy, that is, the more efficient the engine, the more costly the plant and maintenance. Accordingly, to determine what degree of economy will give the best net efficiency for a given set of conditions is a problem requiring considerable knowledge and experience; this is quite evident from the example of ***net efficiency*** given on page 1,632.

The factors which govern the economy of the engine itself, that is, its steam consumption, may be classified under several groups, as:

1. With respect to the steam;
   - *a.* Initial pressure.
   - *b.* Quality of the steam.

c. Degree of expansion.
d. Condensation.
e. Re-evaporation.
f. Leakage.
g. Range of temperature.
h. Back pressure.
i. Compression.
i. Uni-flow distribution.

2. With respect to construction;
a. Cylinder proportions.
b. Clearance.
c. Jackets.
d. Joints.

3. With respect to the method of expanding the steam;
a. Single stage expansion.
b. Multi-stage expansion.
c. Jacketted transfer expansion
(see fig. 1,263 author's system.)

4. With respect to operation;
a. Piston speed.
b. Rotative speed.
c. Wire drawing.
d. Variable load.

5. With respect to the method of governing;
a. Throttle.
b. Cut off.

6. With respect to the piping auxiliary apparatus;
a. Main steam pipe.
b. Reducing valves.
c. Separators.
d. Reheaters.
e. Exhaust pipe.
f. Condenser.

7. Altitude (back pressure).

All the items tabulated above have a direct effect on the steam consumption of the engine, as will now be explained in the order given.

1. The Steam

Initial Pressure.—It has been carelessly and erroneously stated by most writers that *the efficiency varies directly with th initial* pressure.*

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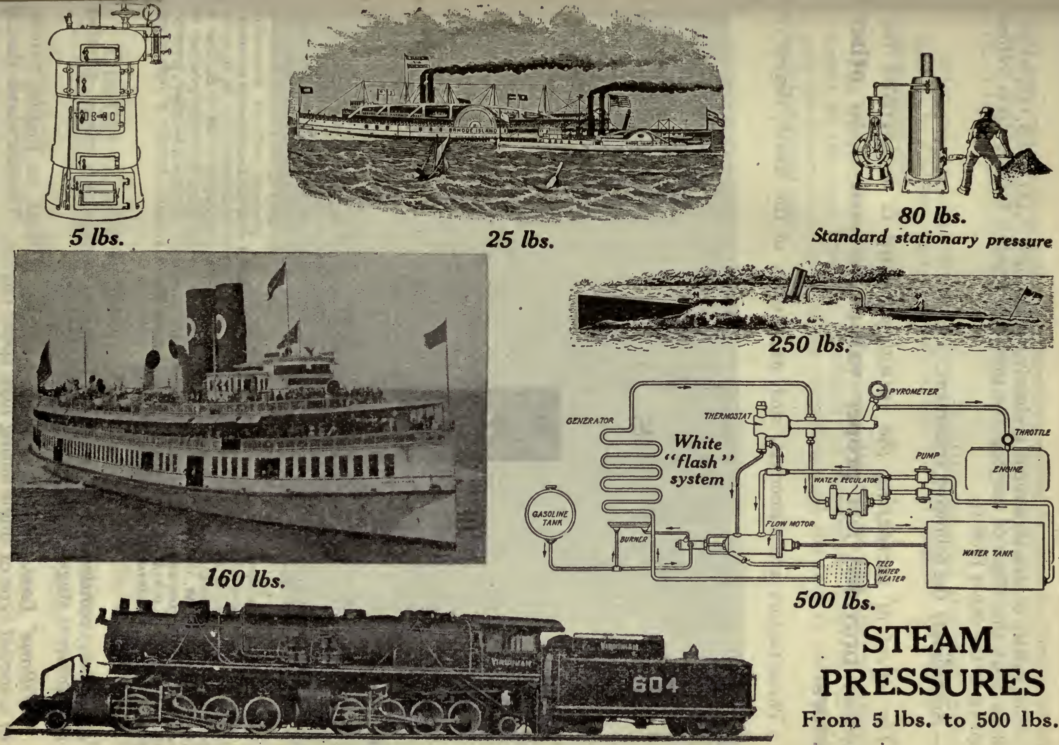
*NOTE.—The economical effect of each factor depends largely upon the other factors that is, any one factor cannot be isolated but must be considered together with all the other con ditions of operation.



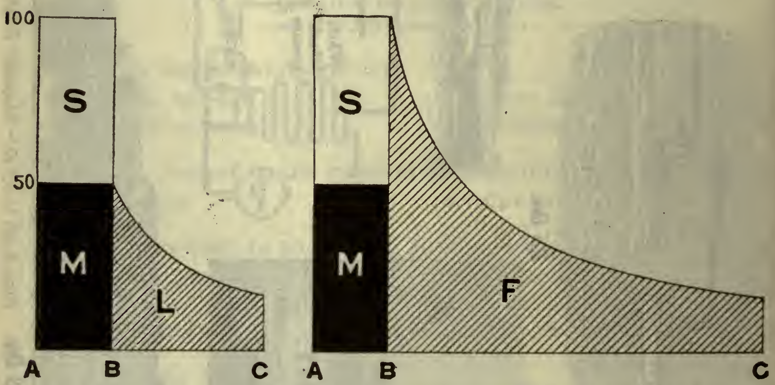
FIGS. 3,134 to 3,140.—Examples illustrating use of steam of various pressures.

The falsity of this statement is clearly seen in figs. 3,141 and 3,142, which show that the effect of varying the pressure depends largely upon values given to the other factors.

Accordingly the following statement:

The effect of increasing the initial pressure is to increase the economy **provided** *that* **proper values** *be given to the* **other factors.**

These factors are numerous, such as, quality of the steam, degree of expansion, ***condensation,*** terminal pressure, temperature range, number of expansion stages, clearance, piston speed, etc.



FIGS. 3,141 and 3,142.—Effect of varying the initial pressure: *the efficiency does* ***not*** *always vary directly with the initial pressure*. In fig. 3,141, assume the card **M,** represent the operation of a steam pump taking steam full stroke of length AB. Suppose the pressure be doubled, the work done and amount of steam used will both be approximately doubled with ***no gain in economy***. Now, considering card **M,** if the stroke be extended to C, and the steam be expanded from B, to C, the work done will be increased by the area **L,** which will increase the economy. Comparing this with simply increasing the pressure without expansion (cards **M** + **S**) it is evident that *the pressure* ***may be increased with no increase*** *in economy*. Again in fig. 3,142, if the pressure be doubled and the steam be expanded from B, to some point C, such as will give an additional card area **F,** greater in proportion to **M** + **S,** than **L** (fig. 3,141) is to **M,** the efficiency will be increased, that is, *increasing the initial pressure in connection with* ***adequate expansion*** *will increase the economy. It should be noted, however, the full theoretical benefit due to expansion* ***is not possible*** *in practice, because of influencing factors such as* ***condensation,*** leakage, clearance, etc.

Willans' Law.—This law is due to P. W. Willans, an English builder of a special type of single acting high speed engine. The law states that *if* *the total steam consumption of a throttling engine be plotted as ordinates on a horse power or mean pressure base, the result is an oblique straight line*, as in fig. 3,143; it is accordingly sometimes called the *straight line law*.

While the law applies especially to throttling engines, it applies approximately to Corliss, and probably other cut off engines up to the most economical load.

The ***most desirable*** initial pressure, depends on the kind of economy aimed at (whether thermal, combined, or commercial economy) and can only be determined by carefully considering all the factors entering into each individual case.

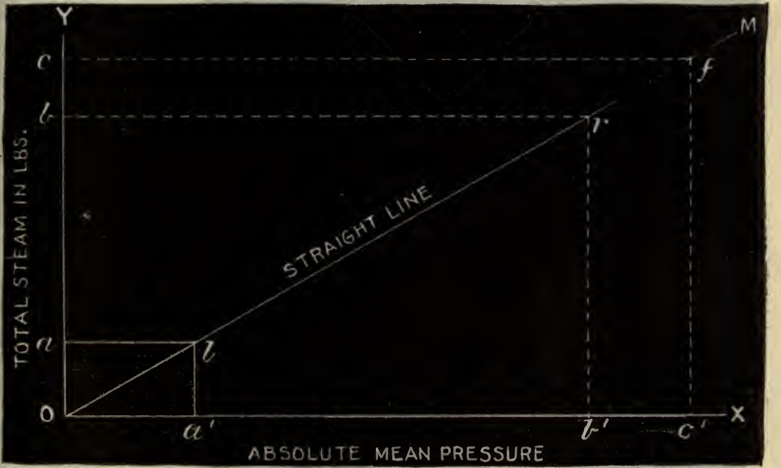


FIG. 3,143.—The ***Willans' straight line law*** for fixed cut off variable pressure (throttling) engines. If the total steam consumption O*a*, O*b*, O*c*, be plotted on the Y, axis, and the corresponding absolute mean pressure O*a*′, O*b*′, O*c*′, on the X axis, obtained from tests with various initial pressures, the then line OM, joining the intersections of the ordinates and abscissæ, at *l*, *r*, *f*, will be a straight line.

Quality of the Steam.—The quality of the steam supplied to an engine is an important consideration, as it has a marked influence on the economy. The steam delivered for a boiler may be, either:

1. Wet.
2. Saturated, or,
3. Superheated.

While the introduction of water in the steam has according to experiment of Carpenter and Marks, no practical influence on the dry steam consumption of the engine, it nevertheless represents so much energy removed from the boiler.

Accordingly with boilers furnishing wet steam, a separator should be used.

The separator placed as near the engine as possible, so this water may be recovered and returned to the boiler by suitable means.*

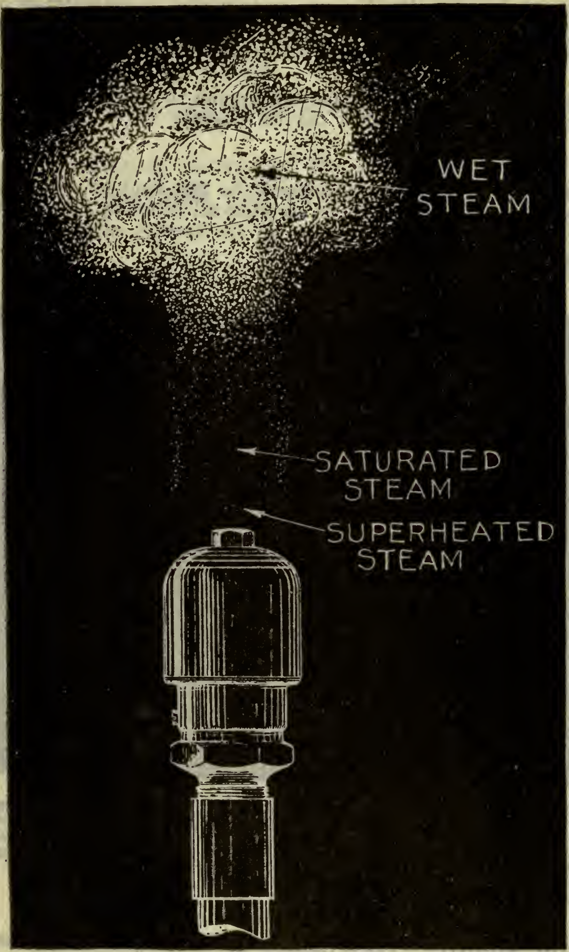
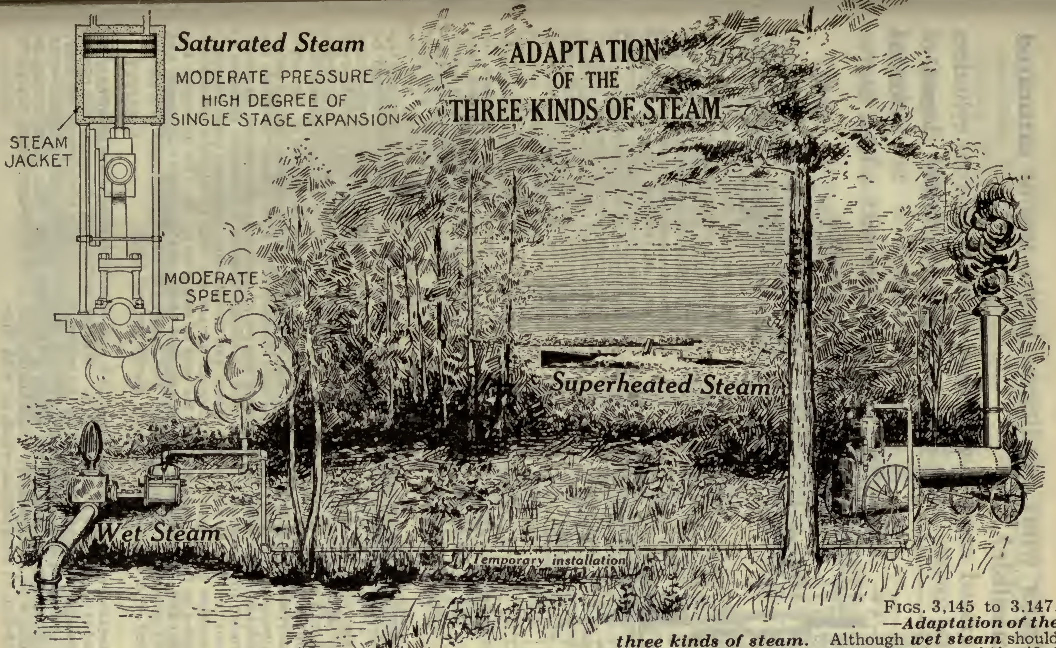


FIG. 3,144.—***Quality of the steam.***—The phenomenon of steam escaping from a boiler illustrates the three kinds of steam supplied to engines: ***1, superheated steam*** is used with high expansion ratios to secure great economy by preventing cylinder condensation; ***2, saturated steam*** is the kind generally used where moderate degree of economy is sought; very good results are obtained in single jacketted cylinders with moderate pressures and high degrees of expansion; ***3, wet steam*** is the kind to be avoided except in special cases (fig. 3,147) but is the kind the engine usually gets, due to improper plant design.



FIGS. 3,145 to 3.147.—*Adaptation of the three kinds of steam.* Although ***wet steam*** should never be used where there is any regard for economy, its use is justified under conditions such as shown above, where a pump is used for a short time to drain an area, or furnish water for sinking piles in bulkhead construction or other rough work of a temporary nature. In such cases the expense of extra fuel due to heat lost between boiler and engine would be less than the cost of properly placing the boiler and insulating the pipe. ***Saturated steam*** is used with good economy with a steam jacketted cylinder when the jacket is used under proper conditions as in the author's system indicated above and on page 1,068 (see also Chapter 52). The high speed launch shown above with its triple expansion engine requires ***superheated steam,*** to secure the maximum power output for a given weight of machinery.

Ques. Under what conditions is the use of saturated steam permissible?

Ans. Saturated steam may be used in non-jacketted cylinders with a moderate degree of expansion or in jacketted cylinders with a high degree of expansion.

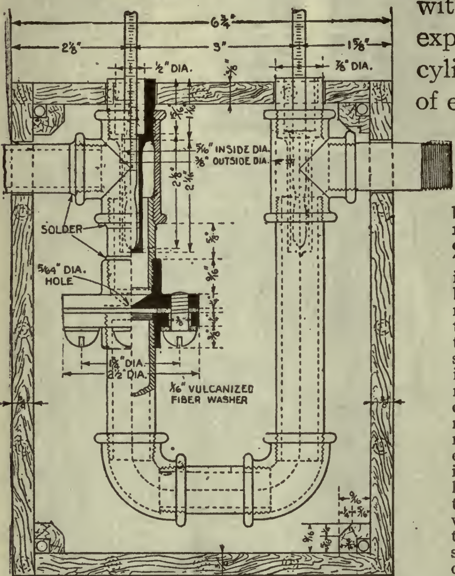


FIG. 3,148.—Throttling calorimeter. *It consists of* ½ inch brass pipe and fittings including two tees, a pair of flanges, an orifice plate between the flanges, and two thermometer cups containing thermometers which are screwed into the tees. The whole is surrounded by suitable insulating material and encased in a box. ***In testing,*** the percentage of moisture is determined by observing the number of degrees of cooling that the thermometer in the low pressure steam shows below the normal reading for dry steam, and dividing that number by the constant number of degrees representing 1 per cent of moisture. To determine the normal reading of the low pressure thermometer corresponding to dry steam, the instrument should be attached to a horizontal steam pipe in such a way that the sampling nozzle projects upward to near the top of the pipe, there being no perforations and the steam entering through the open top of the nozzle. The test should be made when the steam in the pipe is in a quiescent state, and when the steam pressure is maintained constantly at the point observed on the main trial. If the steam pressure fall during the time when the observations are being made, the test should be continued long enough to obtain the effect of an equivalent rise of pressure. To find the constant for 1 per cent of moisture, *divide the latent heat of the steam supplied to the calorimeter at the observed pressure or temperature by the specific heat of superheated steam at atmospheric pressure* (.46) *and divide the quotient by 100.* Finally ascertain the percentage of moisture *by dividing the number of degrees of cooling by the constant,* as above noted. To determine the quantity of steam used by the calorimeter it is usually sufficient to calculate the quantity from the area of the orifice and the absolute pressure, using Napier's formula for the number of lb. which passes through per second; that is, *absolute pressure in lb. per sq. in. divided by* 70 *and multiplied by the area of orifice in sq. in.* To determine the quantity by actual test, a steam hose may be attached to the outlet of the calorimeter and carried to a barrel of water on platform scales. The amount of steam condensed in a certain time is determined, and thereby the quantity discharged per hour.

NOTE.—***In testing engines for economy,*** it is important to determine the ***quality of the steam*** as to the amount of moisture contained in it. Without such a test no proper idea can be obtained of the amount of steam actually used by the engine, to produce a horse power per hour, for the steam may contain a large amount of ineffective moisture in the form of spray—even as much in weight, in an extreme case, as the weight of the steam itself. This quantity may be determined by testing with a calorimeter. Calorimeters may now be bought which are fairly reliable if used according to instructions which go with them.

That is, with unjacketted cylinders the temperature range should be moderate, but with jacketted cylinders the range should be high, the controlling factor being condensation.

Ques. What is the advantage of using superheated steam?

Ans. It reduces or prevents condensation.

Ques. What are the disadvantages?

Ans. Increased difficulty of securing proper lubrication, highness of stuffing boxes, higher first cost and depreciation.

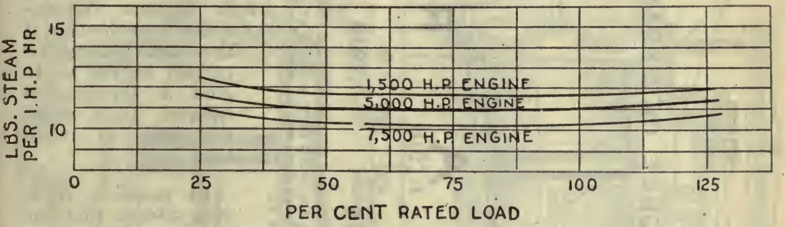


FIG. 3,149.—Economy curves for various sizes of grid iron valve compound condensing engines, variable load with saturated and superheated steam. Boiler pressure, 165 pounds, 100° Fahr. superheat.

Ques. What is the saving due to superheating?

Ans. According to Ripper, the condensation at cut-off is reduced 1 per cent for each 7.5 degrees Fahrenheit of superheat.*

Ques. What conditions favor superheat?

Ans. High degree of expansion, slow speed, constant load, and high fuel cost.

Ques. How much superheat should be given to the steam?

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*NOTE.—The value arrived at is from experiments made by Ripper on a small Schmidt engine cylinder. The figure applies only to their experiments, and they will, of course, be subject to some modification for the numerous types of engines and variable conditions occurring in practice.

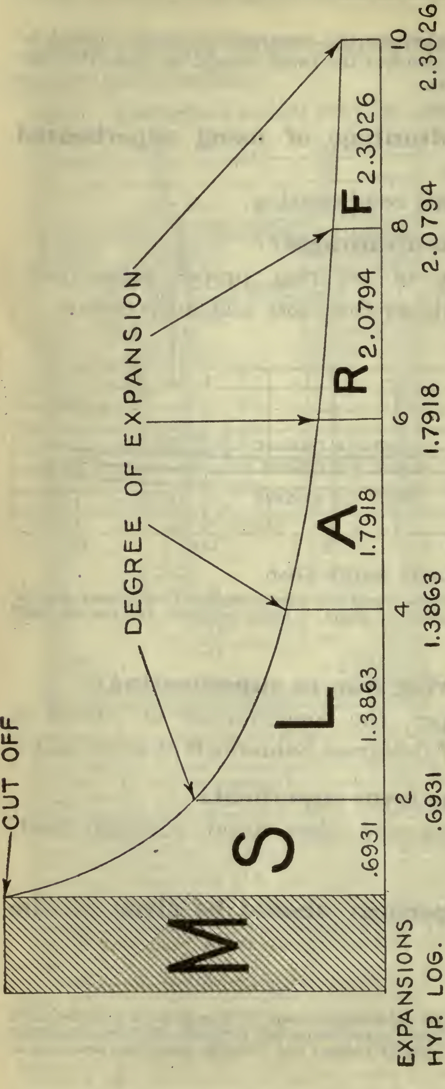


FIG. 3,150.—Diagram illustrating *degree of expansion.* Showing graphically the theoretical increase in work done by a given volume of steam. Thus, if the shaded area **M**, represent the work done without expansion, then if the steam be expanded to 2, 4, 6, 8, and 10 times its initial volume, the theoretical work done will be respectively represented by areas **M+S**, **M+S+L**, **M+S+L+A**, **M+S+L+A+R**, and **M+S+L+A+R+F**.

Ans. For maximum saving possible, the degree of superheating should be such that the steam is exhausted in a saturated state.

Since the object of superheating the steam is to reduce or prevent condensation, evidently if the superheating be carried so far as to give a superheated exhaust this would represent so much loss.

In practice, it is very seldom that the super-heating is carried to the extent of giving a superheated exhaust, in fact the exhaust is, as a rule, not even saturated but wet.

Degree of Expansion.—Theoretically, efficiency *varies directly with the ratio or number of expansions.*

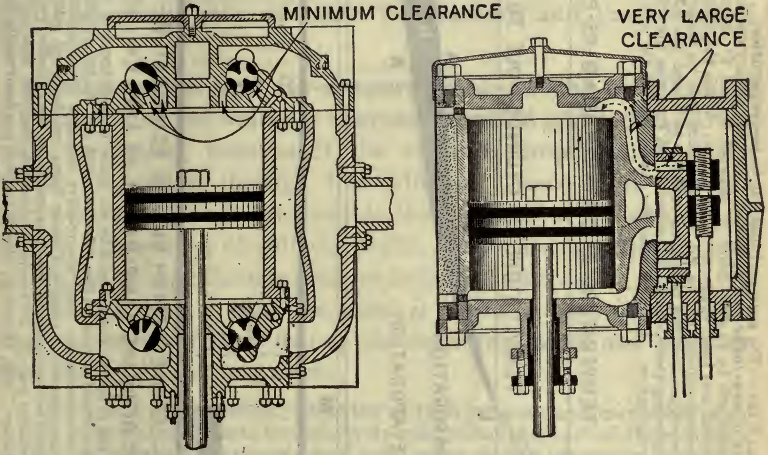
This is illustrated in fig. 3,150. For example, if in the figure the shaded area represent the work done during admission and called unity, then if the steam be expanded to

2 4 6 8 10

volumes, the increase in work done will be respectively equal to the hypobolic logarithm of these numbers or

.6931 1.3863 1.7918 2.0794 2.3026

and the total work done will equal unity plus these values, that is 1 + *hyp. log.* of the number of expansions, hence by expanding the steam the work is increased .6931, 1.3863, etc., fold.



FIGS. 3,151 and 3,152.—Good and poor valve arrangements for high degree of expansion in one cylinder. With the very small clearance in fig. 3,151, a very short cut off may be employed to advantage, but with the arrangement shown in fig. 3,152, the saving due to very short cut off is largely offset by the effect of the excessive clearance which is an inherent defect of the Meyer cut off gear.

Ques. Is this proportion of gain obtained in practice?

Ans. *No.*

Ques. Why?

Ans. Because of influencing factors, such as the quality of the steam, *condensation*, leakage, clearance, action of the valve gear, etc.

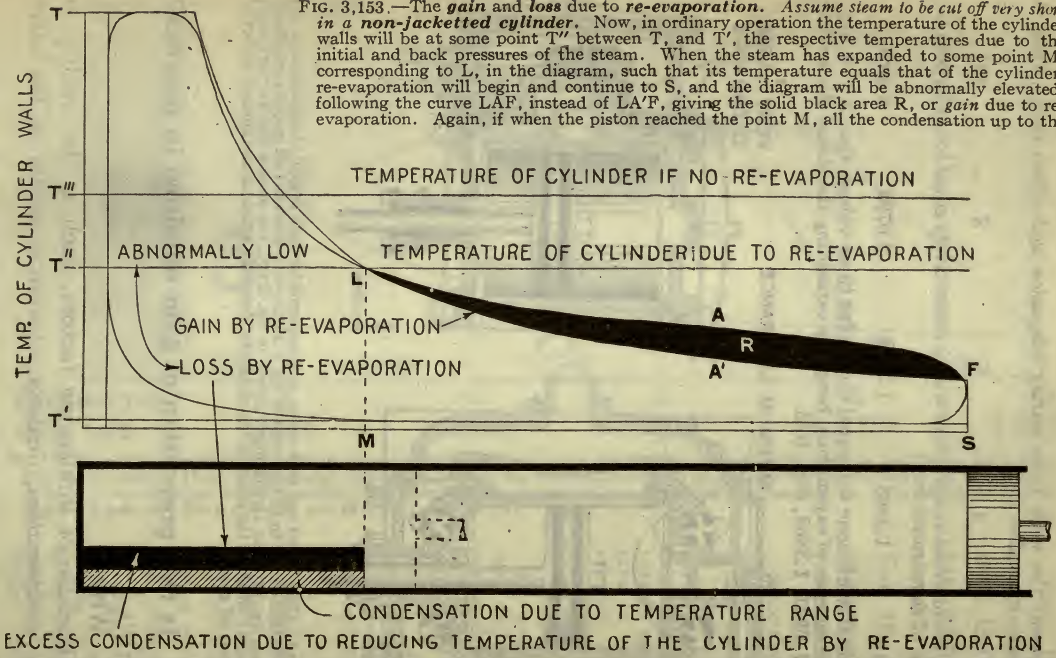

T
T'''
T''
T'
TEMP. OF CYLINDER WALLS
TEMPERATURE OF CYLINDER IF NO RE-EVAPORATION
ABNORMALLY LOW
TEMPERATURE OF CYLINDER DUE TO RE-EVAPORATION
L
A
R
A'
F
GAIN BY RE-EVAPORATION
LOSS BY RE-EVAPORATION
M
S
CONDENSATION DUE TO TEMPERATURE RANGE
EXCESS CONDENSATION DUE TO REDUCING TEMPERATURE OF THE CYLINDER BY RE-EVAPORATION

FIG. 3,153.—The **gain** and **loss** due to ***re-evaporation.*** *Assume steam to be cut off very short* ***in a non-jacketted cylinder.*** Now, in ordinary operation the temperature of the cylinder walls will be at some point T″ between T, and T′, the respective temperatures due to the initial and back pressures of the steam. When the steam has expanded to some point M, corresponding to L, in the diagram, such that its temperature equals that of the cylinder, re-evaporation will begin and continue to S, and the diagram will be abnormally elevated, following the curve LAF, instead of LA′F, giving the solid black area R, or *gain* due to re-evaporation. Again, if when the piston reached the point M, all the condensation up to the point could be *instantly removed* from the cylinder, there would be no water to re-evaporate, thus the latent heat supplied to re-evaporate the water in the first instance would remain in the cylinder metal and its temperature relatively higher as represented by T‴, and the condensation would be considerably less, the relative amounts being indicated by the cross section and solid black areas in the cylinder section which represent a puddle of water lying on the lower side of the cylinder. This solid black section representing excess condensation due to abnormally low temperature of the cylinder *is a* ***loss*** indirectly caused by re-evaporation and since this loss is greater than the gain R, by re-evaporation the resultant effect of re-evaporation, is a loss, hence the misleading expression "*loss by re-evaporation,*" it is, therefore, clearly *loss due to* ***abnormally low*** *temperature of cylinder.*

Ques. How many expansions ordinarily desirable in single cylinder non-jacketted stationary engines?

Ans. *From* ***three*** *to* ***five*** *non-condensing and from* ***five*** *to* ***seven*** *condensing.*

Ques. How many expansions are desirable in single cylinder jacketted engines?

Ans. From ***five*** to ***seven*** non-condensing, and from ***seven*** to ***ten*** condensing, depending on the type of valves and valve gear used, especially with reference to clearance and the action of the gear.

Condensation and Re-evaporation.—In the operation of a steam engine the walls are alternately heated and cooled during each revolution, because of the variable temperatures of the steam in passing through the cylinder. Since these changes occur so rapidly it is not possible for the *metal* of the containing surface (consisting of cylinder walls, heads, steam passages and piston) to change its temperature in like manner.

What happens is this: the *metal is heated to some intermediate temperature*—***less*** *than that of the incoming steam and* ***greater*** *than that of the outgoing steam.*

The result is that steam, admitted at high pressure and correspondingly high temperature, meets the relatively cold metal and part of it condenses, this condensation continues until some point during expansion is reached where the temperature of the steam (because of the decrease in pressure) is reduced to that of the cylinder metal.

A further reduction in pressure (due to continued expansion and drop) causes the temperature of the steam to become less than that of the metal, accordingly part of the condensate is *re-evaporated.*

While this re-evaporation effects a small gain by elevating the expansion line, it is more than offset by the heat taken from the metal, that is, the resultant effect is a loss.* In other words, the reduction in temperature of the metal due to evaporation causes more loss by condensation than is gained by the elevation of the expansion line on the indicator card.

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*NOTE.—The term ***loss*** *by re-evaporation* is used so frequently by writers, that no doubt, a wrong impression is conveyed to most readers. It should be distinctly understood that re-evaporation causes a gain by increasing the area of the indicator cord; it is the ***cost*** *of obtaining this* ***gain*** (*i.e.*, robbing the cylinder metal of heat) that *offsets this* ***gain*** by a ***greater loss.***

Ques. Ordinarily how much condensation occurs in single unjacketted cylinders using saturated steam?

Ans. From 10 to 60 per cent.

The following table is from the catalogue of the Buckeye Engine Co., builders of "automatic" cut off engines with riding cut off and special valve gear. The table is based on an initial pressure of 80 lbs. and different points of cut off with Buckeye Engines and others having similar clearance:

Water Consumption and Condensation Loss at Various Cut Offs

Cut off in % of stroke	**M.E.P.**	Absolute terminal pressure	***Indicated Rate*** Lbs. of water per I.H.P. per hour	***Assumed Rate*** Actual rate	***Assumed Rate*** % Loss	Product of columns 1 and 6
.1	18	11	20	32	58	5.8
.15	27	15	19	27	41	6.15
.2	35	20	19	25	31.5	6.3
.25	42	25	20	25	25	6.25
.3	48	30	20	24	21.8	6.54
.35	53	35	21	25	19	6.65
.4	57	38	22	26	16.7	6.68
.45	61	43	23	27	15	6.75
.5	64	48	24	27	13.6	6.8

It appears from the table that while the best indicated economy is when the cut off is from 15 to 20% of the stroke, the most economical cut off, considering the un-indicated loss, is at 30% of the stroke. The last column shows that the actual amount of cylinder condensation is practically a constant percentage of the cylinder volume up to the point of cut off, increasing only for 5.8% at 1-10 cut off to 6.8% at ½ cut off.

NOTE.—As expressed by Rankine: "Conduction of heat to and from the metal of the cylinder, or to and from liquid water contained in the cylinder, has the effect of lowering the pressure at the beginning and raising it at the end of the stroke, the lowering effect being on the whole greater than the raising effect. In some experiments the quality of steam wasted through alternately liquefaction and evaporation in the cylinder has been found to be greater than the quantity which performed the work."

Temperature Range.—*When the temperature of the steam is greater or less than that of the cylinder, a transfer of heat takes place respectively* **from** *or* **to** *the steam.*

Experiments show that this transfer takes place at the rate of *20 to 25 B.t.u. per square foot of the cylinder metal per degree of temperature difference per minute.*

Accordingly *the greater the temperature difference, the greater the losses due to condensation and indirectly to re-evaporation*, and due to these effects steam can be expanded *only* to a limited degree in a single cylinder without incurring losses that more than offset the gain due to expansion.

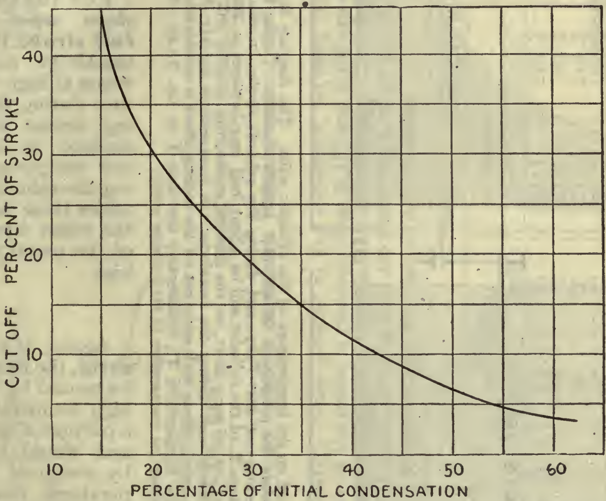
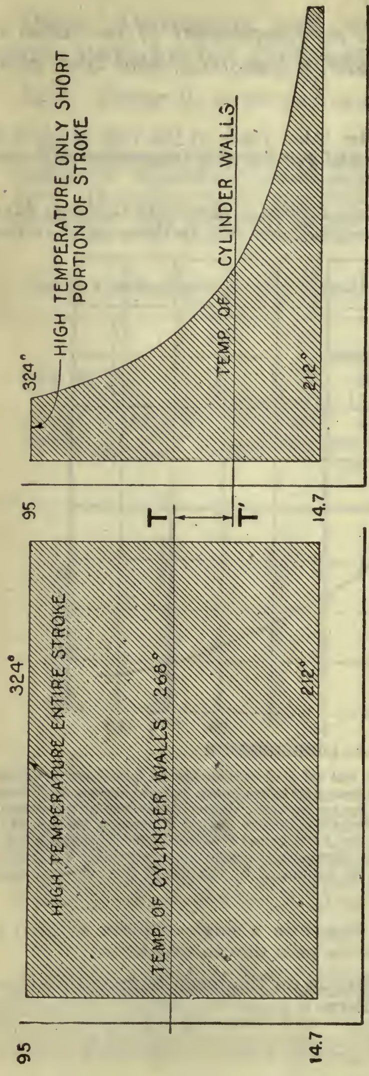


FIG. 3,154.—Diagram showing relation between cut off and percentage of initial condensation. It has been found by experiment that heat transfer takes place to and from the cylinder walls at the rate of from 20 to 25 *B.t.u.* per square foot per degree difference of temperature per minute (*Marks.*) Since most of the condensation occurs during admission the quality of the steam at cut off is a measure of the economy. This may be ascertained by comparison of the indicator diagram with the experimentally determined steam consumption. The quality of the steam at cut off and economy are directly influenced by the size of the cylinder, speed, and number of expansions.

Cylinder condensation depends directly on the mean temperature of the cylinder metal; this in turn depends on:

1. Temperature range, and
2. Cut off.


95
324°
HIGH TEMPERATURE ENTIRE STROKE
TEMP OF CYLINDER WALLS 268°
212°
14.7
T
T'
95
324°
HIGH TEMPERATURE ONLY SHORT PORTION OF STROKE
TEMP. OF CYLINDER WALLS
212°
14.7

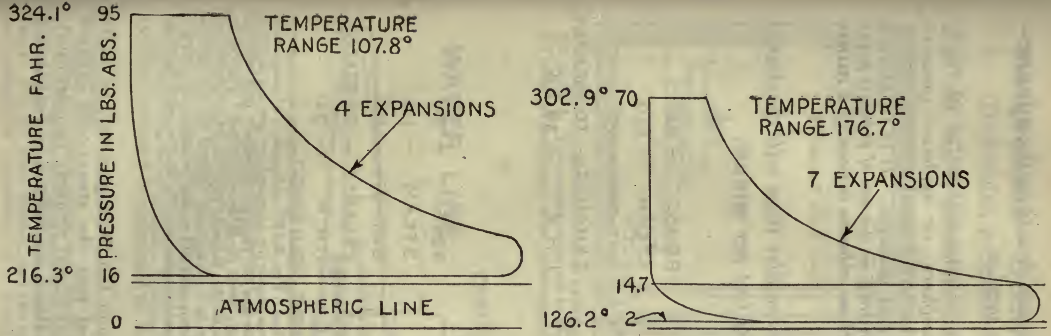
FIGS. 3,155 and 3,156.—Effect of cut off on the temperature of the cylinder walls. The diagram fig. 3,155 represents the operation of a steam pump working between the pressure of 95 and 14.7 pounds absolute. During the forward stroke the cylinder is exposed to steam at a temperature of 324° Fahr., and during the exhaust stroke, to steam at a temperature of 212°. Under such conditions the temperature **T**, of the cylinder will be about half way between or 268°. Now, if an engine work with the same *temperature range*, but cut off very short, on the forward stroke, the cylinder will be exposed to the high temperatures only a short part of the stroke and during the remainder the temperature quickly drops. On the return stroke the conditions are the same in both cases. Since with early cut off the cooling effect is considerably more marked, it is evident that the cylinder would assume some temperature **T'**, considerably less than **T**, in fig. 3,155. Hence for equal temperature ranges, ***the shorter*** *the cut off the* ***lower*** *the temperature of the cylinder walls.*

It must be evident that for a given temperature range then *the mean temperature of the cylinder metal* ***may vary largely,*** thus affecting the condensation correspondingly.

For instance, *if steam were admitted* ***full stroke*** the metal would be heated by steam at high temperature during the entering stroke and only exposed to steam of low temperature during the exhaust stroke; under these conditions the mean temperature of the metal would be high.

Again, *if cut off be* ***early,*** the metal would be heated by steam at high temperature only a portion of the stroke, and would be cooled by steam of low temperature during the rest of the stroke and during exhaust, resulting in a relatively low temperature of the metal.

It should also be noted that the mean temperature of the metal may remain constant for various temperature ranges, in which case the



FIGS. 3,157 and 3,158.—Temperature ranges ordinarily employed in stationary engines. Fig. 3,157, non-condensing, ¼ cut off; fig. 3,158, condensing, 1/7 cut off.

condensation may vary considerably, being greater as the difference between the initial temperature of the steam and mean temperature of the metal is greater. Figs. 3,157 and 3,158 are examples of temperature ranges in ordinary stationary practice for single non-jacketted cylinders. The following table gives some temperature ranges and results obtained, showing that *the temperature range* **alone is no index** *of economy:*

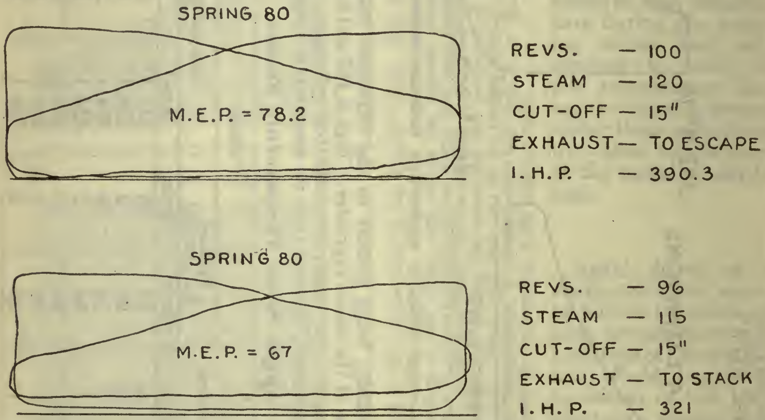
Single Stage Temperature Ranges

Type of engine	Cut-of	Steam				Range of temperature degrees Fahr.	Steam per horse power per hour lbs.
		Pressure lbs. absolute		Temperature			
		Initial	Back	Initial	Back		
Slide valve	$\frac{6}{10}$	95	16	324.1°	216.3	107.8	50
Slide valve	$\frac{6}{10}$	95	2	324.1°	126.2	190.1	40
Automatic cut-off	$\frac{1}{4}$	95	16	324.1°	216.3	107.8	32
Automatic cut-off	$\frac{1}{6}$	95	2	324.1°	126.2	190.1	26
Corliss....................	$\frac{1}{4}$	95	16	324.1°	216.3	107.8	25
Corliss....................	$\frac{1}{7}$	95	2	324.1°	126.2	190.1	20
Corliss....................	$\frac{1}{7}$	95	2	424.1	126.2	297.9	18

Back Pressure.—This is *the pressure which acts on the* ***advancing*** *face of the piston and* ***opposes*** *its motion.*

In the operation of a non-condensing or condensing engine the back pressure is normally higher than the atmospheric, or condenser pressure respectively. This difference of pressure is due to frictional resistance which the steam encounters in the exhaust pipe and passages. *It is in practice nearly always* ***too great*** *due to* ***false economy*** *in the use of exhaust pipes too small, together with a* ***multiplicity of close elbows*** *instead of a* ***minimum number of easy bends.***

In no case should the exhaust pipe be smaller than will discharge the steam at a velocity of ***not over*** *4,000 feet per minute.*



FIGS. 3,159 and 3,160.—Two diagrams from a non-condensing steam lighter engine showing excessive back pressure due to poorly designed exhaust line. Fig. 3,159, exhaust to atmosphere; fig. 3,160 exhaust to stack. The excess back pressures above the atmosphere are for the two cases, 5 and 10 pounds respectively. ***There is no excuse*** *for such performance*, and radical changes should be made in the exhaust line where any such cards are obtained. The cards are a striking example of very bad practice. No doubt the exhaust pipe was too small and possibly there was insufficient exhaust port opening in the engines. Added to this, a few close elbows, a three way cock with less than full opening and the excess pressure required for the blast in the stack, account for the high back pressure, however, the coal wasted because of such conditions is not usually considered.

It is not always the fault of the exhaust pipe but the engine may be badly proportioned and it should be ascertained that the steam and exhaust passages forming a part of the cylinder have been properly proportioned, especially in single valve engines where the steam is admitted and exhausted through the same passage.*

Compression.—The reasons for compression are:

1. To furnish a cushion of gradually increasing resistance in order to bring the moving parts to rest and change the direction

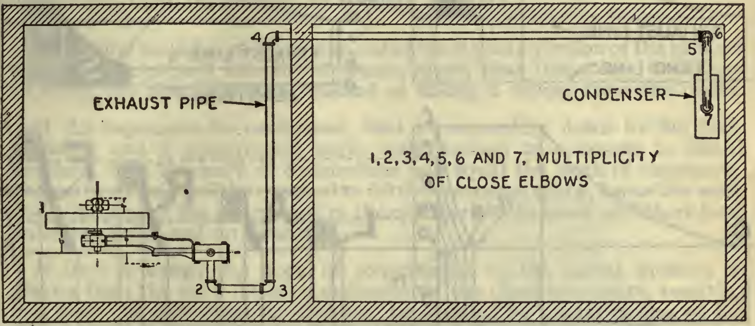


FIG. 3,161.—An arrangement of exhaust piping which was expected to give a 24 inch vacuum, ***but didn't.*** The results obtained in this particular case was 12 to 15 inches. The only reason for this arrangement were *convenience*, and *ignorance* of conditions affecting back pressure.

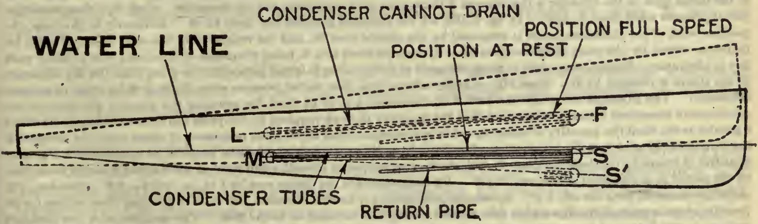


FIG. 3,162.—Another arrangement of exhaust piping which was expected to give a 24 inch vacuum, ***but didn't.*** **MS**, is a keel condenser on a small boat; it was attached to the hull in a horizontal position, that is, parallel to the water line with the idea that, in this position, it would offer least resistance to the motion of the boat. When the boat was running full speed, the stern settled so that the condenser became inclined to some position as **LF**. Since the "return" to air pump was at the forward end, the condenser would not drain, hence there was little or no vacuum. Satisfactory results were not obtained until the condenser was pitched downward as represented by the dotted line **MS′**.

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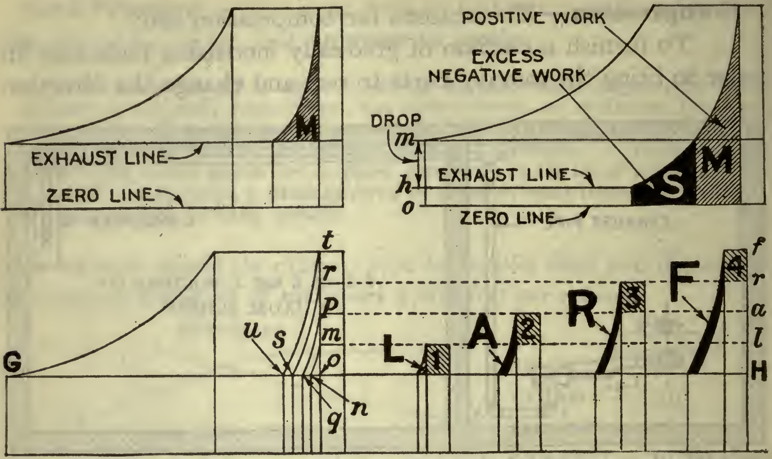
*NOTE.—The importance of this must be apparent when it is considered that some engines operate with mean effective pressures as low as 20 pounds. In such case, if the back pressure be unnecessarily elevated 1 pound, the loss is 1-20 or 5%. Figure 5% of the cost of coal per year, or 1% for that matter, and the result will illustrate how important it is to reduce the back pressure to a minimum by a direct and correctly proportioned exhaust line.



Figs. 3,163 to 3,165.—Diagrams, illustrating economic effect of compression *Case I.—Compressing to initial pressure when the expansion is continuous* (*i. e.*, no drop) In fig. 3,163, the work of compression measured by the shaded area M, is equal to the work of expansion of the clearance steam measured by the same shaded area M, thus the work and efficiency of the steam used in the cylinder is the same as if there were no compression and no clearance *Case II.—Compressing to initial pressure when there is drop.* In fig. 3,165, let the terminal pressure *om*, drop to *oh* Evidently in this case, the positive work of the clearance steam is measured by the shaded area M, and the resistance of the work of compression for clearance steam by the more extensive area M + S, the solid black area S, being the excess of resistance over positive work due to clearance. Hence, *when there is drop, the work of compressing to* initial pressure is so great that the net efficiency of the steam is reduced by it. *Case III.—Compressing to less than initial pressure* when there is *drop* (case of greatest efficiency) The principle on which the period of compression is to be fixed for ensuring the greatest efficiency of the whole steam consumed is based on the fact that the quantity of work required to compress the exhaust steam increases in a greater ratio, than the quantity of steam saved by compression; that is, *for equal increments of steam saved, the corresponding increments of work expended increase continually.* In other words it is useless and wasteful of work to save a greater quantity of steam by compression than that of which each increment severally will yield at least as much useful work done on the piston as is expended in compressing it ***To demonstrate,*** let piston area be 1 sq. in.; stroke, 6 ft.; clearance, 7%; initial pressure, 63 lbs.; cut off 2 ft By simple calculation for these conditions, the ratio of the numerical values for *the quantity of steam expended to the net work done without compression is* found to be **(1:.928)** This proportion is the standard ratio with which that of the corresponding values for compressed steam is to be compared, under the given conditions In fig. 3,163 let the exhaust be closed progressively earlier so that steam may be compressed to pressures indicated by the elevations *l*, *a*, *r* and *f*; being equal increments of pressure. Now since the portion of the resisting area below the line GH, remains unchanged whether there be compression or not, the only portions of the works of compression that need be brought into calculation in the present argument are the extra portions—the triangular sections lying above the exhaust line namely, *mno*, *pqo*, *rso* *tuo*: Now, the successive ***differences*** *of these areas*, or increments of work areas, *represent the increase of works required to produce the successive* ***equal*** *increments of pressure in the clearance space*, these successive ***differences*** *or increments of work areas* being projected and shown separately for clearness by the shaded and solid black areas L, A, R, F ***By calculation,*** *the ratio of the steam expended and the net work done during the stroke are:* for L, 1 to 175; for A 1 to 495; for R, 1 to 701 for F, 1 to .896, and as previously obtained without compression **(1 to .928)** ***At length,*** it appears that for each increment reclaimed, the increment of compressive work as (L. A. R or F), even the last and greatest increment F, is less comparatively than the corresponding increment of net work done by the steam on the piston (as *l*, *a*, *r*, or *f*) which in the case of F, and f, is in the ratio .896 to .928. From this it would appear that the exhaust steam might with economic advantage be compressed in the clearance space to initial pressure: Not so, for by dividing Hf, into more minute increments, say 10, it is found that by compressing to about .85 of Hf, the highest efficiency is obtained under the conditions assumed. The calculations for the above figures are to be found in "*The Steam Engine,*" by D. K Clark (Part 4, pages 390 to 399) being presented in great detail.

of the force acting upon them without the shock which would follow upon the sudden opening of the valve.

2. To improve the economy by filling the clearance space with steam that otherwise would be wasted.

By closing the exhaust before the end of the stroke a portion of the exhaust steam is compressed into the clearance space, thus this steam is utilized in filling the clearance space instead of taking a supply from the boiler.

If the expansion be *continuous*, that is, expanding down to the back pressure, and if compression begin early enough to compress to initial pressure, then the work of compression will equal the work of expansion, and no live steam will be required to fill the clearance space, hence the work and efficiency of the steam used in the cylinders is the same as if there were no compression and no clearance.

If there be **drop,** the work of compression to the initial pressure is greater than the work done by expansion of the clearance steam, resulting in a loss. In such case, the loss may be avoided by a later closing of the exhaust valve compressing the steam to *less* than initial pressure.

The general opinion prevails that the greatest saving due to compression is obtained by compressing to initial pressure whether there be drop or no drop, however, in all cases the best results are obtained by compressing to a pressure *less* than the initial pressure, as illustrated in figs. 3,163 to 3,165.*

In practice, re-evaporation during the first part of compression will have considerable influence upon compression, raising the pressure at various points above normal.†

In engines not having independently driven exhaust valves, and varying the cut off by the method of combined variable angular advance and variable throw, the compression is increased as the cut off is shortened, moreover, the exhaust becomes cramped due to the decreasing travel of the valve.

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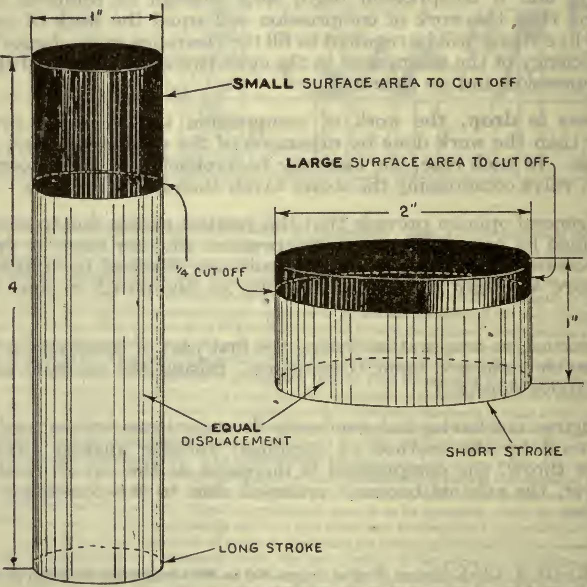
*NOTE.—D. K. Clark (Steam Engine, pages 390 to 399) treats this problem in great detail and should be read by all interested in the subject.

† NOTE.—According to Professor Peabody "the only valid objection to Hirn's analysis is directed against the assumption of dry steam at compression," and this is a very good objection for it is not reasonable to assume, for an engine supplied with saturated steam and running with a reasonable degree of expansion, that the steam will be dry when the exhaust valve closes. In fact Professor Carpenter says (*Trans A. S. M. E.* vol. xii, p. 811) that the steam exhausted from the *h.p.* cylinder of a compound engine showed 12 to 14% of moisture. It must be evident, that since the condensation increases as the cut off is shortened, the greater will be the amount of condensate subject to re-evaporation, and if sufficiently great, re-evaporation will continue after the exhaust valve closes and until the temperature of the mixture has reached that of the cylinder walls.

The excess compression and back pressure thus produced at early cut off, offsets somewhat the economy which otherwise would be obtained.

2. Engine Construction

Cylinder Proportion.—The dimensions to be considered here are chiefly the ***stroke*** *and* ***diameter***, *especially the* ***ratio*** *between the two*.



FIGS. 3,166 and 3,167.—***Cylinder proportion;*** diagram showing relation between displacement and internal surface. Fig. 3,166, ***long stroke;*** fig. 3,167, ***short stroke.*** With the proportions given above it can be shown mathematically that if cut off take place at ¼ stroke in each cylinder, the actual surface of the cylinder enclosing the steam up to cut off is 5.56 square feet in the long stroke engine, and 75% more in the short stroke engine. Accordingly, if in a ***non-jacketted*** cylinder condensation be directly proportional to surface, the condensation in the short stroke engine will be 75% greater than in the long stroke engine when cut off takes place at ¼ stroke in each cylinder. It should be noted that as the cut off is made later the respective surface areas become more and more nearly equal, and if steam be admitted full stroke the surface is 7% greater in the long stroke cylinder. Thus, the loss by condensation in the two cylinders would be more nearly equal as the cut off is lengthened.

While the economic effect is considerably influenced by the ratio, there are mechanical conditions which have to be considered in determining what ratio to use.

With respect to the ratio (stroke to diameter), engines are divided into three classes:

1. Short stroke, or high speed.*
2. Medium stroke, or medium speed.
3. Long stroke, or low speed.

Thus, for a direct connected set, requiring to be run at high speed, such as an engine coupled to a dynamo, or centrifugal pump, a short stroke engine would be used. When a more moderate speed is permissible, a medium stroke engine would be used.

Because of the ***limitations** of **releasing** valve gears*, the revolutions per minute must be relatively low, hence engines with such gears have long strokes.

With respect to the loss by condensation in non-jacketted cyclinder, *the **shorter** the stroke in proportion to the diameter the **greater** the efficiency.*

Other conditions tending to offset this are: 1, greater clearance; 2, greater length of piston circumference past which leakage is liable to occur.

Steam Jackets.—There has been much unnecessary discussion and difference of opinion in regard to the economic value of steam jackets.

An opinion of the absolute value of steam jacketting should not be formed by considering only a single set of tests on the benefit of jacketing, as often influencing factors are present which offset more or less the saving that would otherwise be possible.

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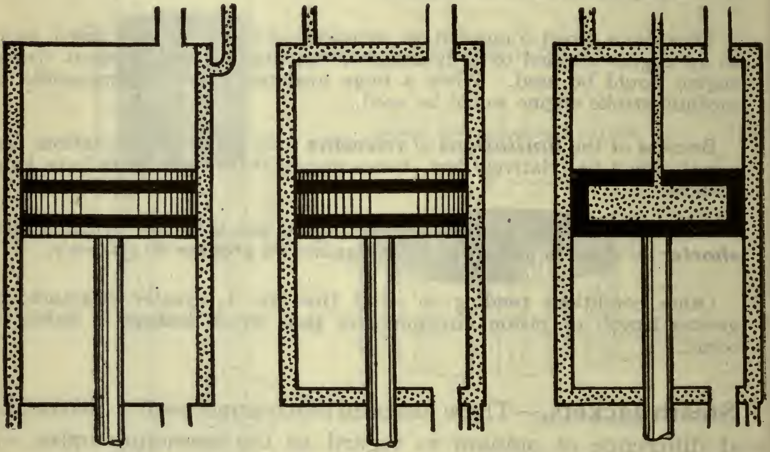
*NOTE.—Speed here means rotary speed or *r.p.m.* and not piston speed, in fact in some cases the piston speed of a low speed engine may be higher than that of a high speed engine. The English term for high speed is *quick revolution* and this is more explicit than the American expression.

Thus, though Donkin obtained a saving of 40% by jacketting, the engine had a riding cut off valve, the considerable clearance of which, taken in connection with the short cut off diminished the saving otherwise possible.

The object of jacketting being *to* ***prevent*** *condensation*, it must be evident that *the chief favorable condition is a* ***high degree of expansion*** *in the cylinder*.

To secure full benefit of a jacket, *thorough drainage and avoidance of air pockets is important*.



FIGS. 3,168 to 3,170.—Three degrees of cylinder jacketting. **1**, fig. 3,168, cylinder only jacketted; **2**, fig. 3,169, cylinder and heads jacketted; **3**, fig. 3,170, cylinder, heads and piston jacketted, this arrangement being called *full jacketted*.

The jacket condensate should be returned to the boiler, thus saving the heat it contains.

The aim of jacketting being the same as superheating, the two should not be used together, that is, jackets should be employed with saturated steam only.

The efficiency may be increased by using a higher steam pressure in the jacket than in the cylinder, when the heating surface is not enough to obtain saturated steam during exhaust.

Jacketting may be partial or complete, as shown in figs. 3,168 to 3,170 that is, jackets may be provided for: 1, cylinder alone; 2, cylinder and heads, or; 3, cylinder, heads, and piston, the latter case being called *"full jacketted."*

Ques. Is jacketting more efficient on a short or long stroke engine?

Ans. The efficiency is greater with a long stroke.

Ques. Why?

Ans. Because the ratio of the jacketted surfaces to the cylinder volume is greater.*

Joints.—This term is used to indicate various constructions designed to prevent leakage of steam. There are two general classes:

1. Stationary joints.
2. Sliding contact joints.

Stationary Joints are those provided for the cylinder heads, valve chest covers, etc., made tight usually by gaskets and in some cases by metal to metal contact ground surfaces.

An important point in respect to such joints is that there be a sufficient number of bolts to maintain them tight, otherwise there is constant loss by leakage.

In a well designed engine, the bolts should be so proportioned that the maximum stress on each bolt (figured to bottom of thread) should not exceed 5,000 pounds per square inch of cross section.

Sliding Contact Joints comprise the contact surfaces: 1, between valve and seat; 2, between piston rim and cylinder barrel; 3, between valve, stem, piston rod, and the stuffing boxes.

In each case it must be evident that the part should be of proper design to prevent leakage, as numerous small losses possible at the joints, would in the aggregate, considerably increase the steam consumption, thus impairing the efficiency.

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*NOTE.—With non-jacketted cylinders, the earlier the cut off, the more efficient the long stroke engine in regard to condensation, because comparing short and long stroke engines of equal cylinder volumes, when steam is cut off short, it is during admission subjected to less metal surface. As the cut off is lengthened the metal surface volume ratio for short and long stroke engines becomes nearer equal (being greater in the long stroke cylinder when steam is admitted to end of stroke) which indicates that ***late cut offs** are more desirable for **short stroke*** engines.

Uni-flow Engines.—The introduction of a separate exhaust port remote from the admission port, is the means employed in this engine to increase the economy.*

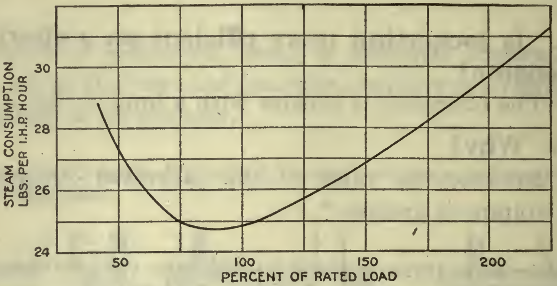

FIG. 3,171.—Steam consumption curve of Stumpf uni-flow engine ***non-condensing*** using ***saturated steam.***

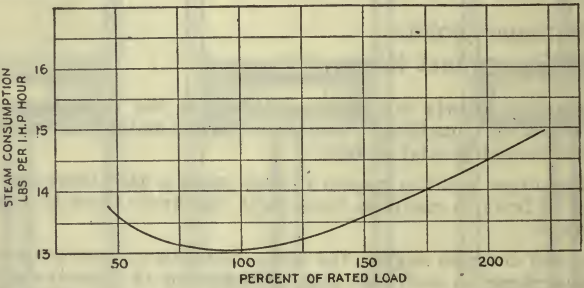

FIG. 3,172.—Steam consumption curve of Stumpf uni-flow engine ***condensing*** using ***saturated steam.***

In general, the efficiency of non-condensing, non-jacketted uni-flow engines is about equal to that of the best compounds.

A difficulty inherent in the working principle of uni-flow engines is compression. Because of the early closing of the exhaust ports various expedients are employed to prevent excessive compression, especially when operating non-condensing. Because of the early compression, the engine is best suited for condensing operation.

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*NOTE.—The *uni-flow engine* is treated at considerable length in a separate chapter. See pages 591 to 601.

NOTE.—***Saturated steam consumption*** of 19.49 lbs. is claimed by the manufacturer for a Skinner 15×16 uni-flow engine non-condensing, operating at half load, steam pressure 127 lbs.

3. Method of Expanding the Steam

Single and Multi-Stage Expansion.—Where the degree of economy aimed at is higher than that obtainable with 4 or 5 expansions non-condensing, or 6 to 8, condensing, *the expansions should be divided into two or more* ***stages*** *by means of additional cylinders, thus* ***dividing*** *the temperature range so that it will not be excessive in any one cylinder.*

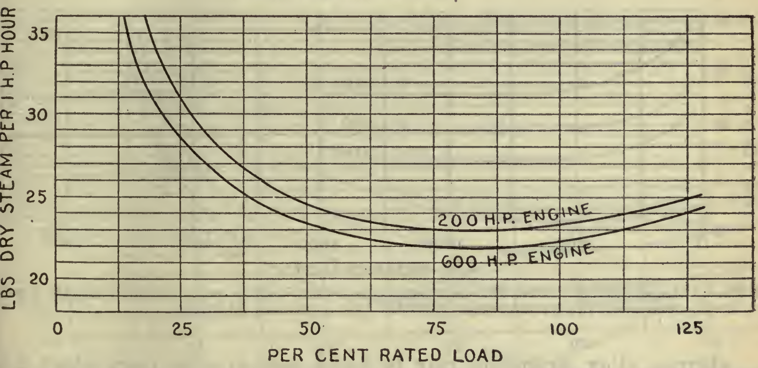


FIG. 3,173.—Economy curves for various sizes of simple non-condensing four valve engine, variable load, 130 pounds boiler pressure.

By this means condensation is greatly reduced. The saving that may be expected by the use of multi-stage expansion, condensing or non-condensing, as compared with single stage expansion, under like conditions, may be roughly stated as from 10 to 50%, varying with conditions, management, etc., so that even though the first cost of the plant will of necessity be considerably more per horse power, the larger investment is generally justified.

To determine with precision whether such investment be advisable is a matter requiring expert advice, and all the conditions entering into the problem should be very thoroughly considered.

For multi-stage expansion, the highest economy is obtained in plants where the load of the engine is uniform (such as in water works pumping engine, marine engine, etc.), because the engine may be so designed that the power corresponding to the working load will be developed with the most economical number of expansions.

Uniform load, extreme ratios of expansion, great care to avoid leakage; 1, of steam through the joints; 2, of heat through covering of pipes and cylinders, and size of areas for passage of steam to and from the cylinders, all have great influence in fixing the possible economy of the engine; moreover, to obtain best results, expert management of both engine and boiler must be added.

In examining the catalogues of manufacturers, common sense will indicate to the reader, that the cylinder proportions adopted are those which are best adapted to be used in making standard patterns for a line of engines, whereas, other dimensions may be better used in some cases where the highest economy attainable is desired, with comparative disregard of first cost.

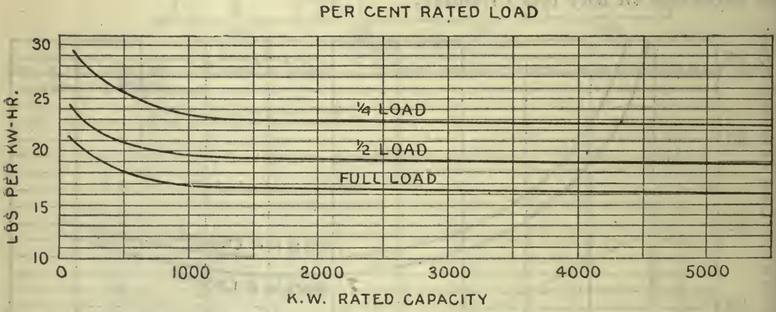


FIG. 3,174.—Economy curves for compound condensing piston valve engine variable load. Boiler pressure 177½ pounds; superheat 90° Fahr.; vacuum 26 inches.

4. Operation

Piston Speed.—For constant speed of rotation, the piston speed may be increased by lengthening the stroke.

In increasing the piston speed in this manner, the absolute value of such increase is to improve the economy, because a greater weight of steam passes through the cylinder per minute, resulting in reducing the percentage of condensation.

Rotative Speed.—With non-jacketted cylinders, the effect of increasing the rotative speed is to reduce the cylinder condensation, but the efficiency is not usually increased by this means, because of the poor steam distribution due to the types of valve gear available for such service.

For jacketted cylinders it is generally agreed that *the* ***slower*** *the rotative speed the* ***more efficient*** *the jacket* because of the longer time in which each charge of steam can receive heat transmitted from the jacket steam through the cylinder walls.

High rotative speed means ***short stroke*** *and* ***large clearance.***

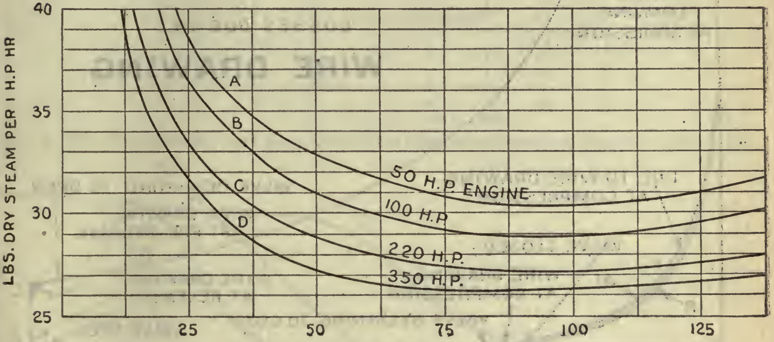


FIG. 3,175.—Economy curves for various sizes of simple non-condensing high speed piston valve engine variable load. Boiler pressure 125 pounds.

Wire Drawing.—Since valves *cannot close or open* ***instantly,*** *the gradual closing or opening of the port causes the pressure to be* ***less*** *than it should be*, owing to the frictional resistance encountered. Accordingly, the "corners" of the diagram are rounded instead of sharp, the card area thus lost reducing the diagram factor in proportion as shown in fig. 3,176.

The extent of this loss depends on the velocity of the valve in opening and closing, and various methods have been employed in designing valve gears to obtain a quick opening and closing of the valve, conspicuous examples being the releasing gears, and riding cut off gears.

The loss due to wire drawing is most marked in engines of the so called "automatic cut off" type which vary the cut off by the method of *combined variable angular advance and variable throw.* In these, not only is wire drawing caused by the slow opening and closing of the valve, but is considerably increased by the reduction of port opening at early cut off caused by decreased throw. The same thing may be said of engines (as for instance locomotives), using the shifting link motion as a variable expansion gear.

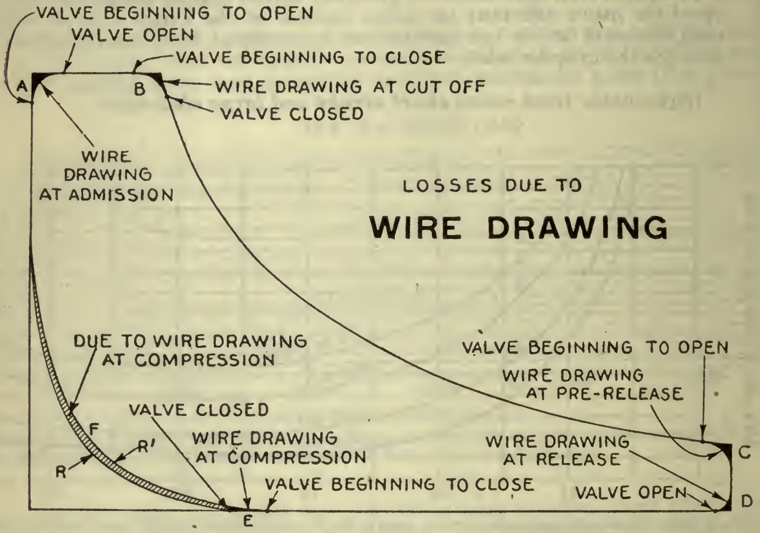


FIG. 3,176.—Action of the valve gear in reducing the area of the diagram by wire drawing. While the steam valve is opening for admission, the solid black area A, is lost; while closing for cut off area, area B, is lost; while the exhaust valve is opening for pre-release and release, areas C, and D, are lost; while closing for compression area E, is lost, the effect of which is to elevate the entire compression curve from R, to R′, taking away from the diagram the area F. It is evident from the figure that if engine valves could *open and close **instantly*** the efficiency would be considerably increased. Such operation in practice is of course impossible.

In a well designed engine using such means for varying the cut off, the port opening for full gear (latest cut off) should be made as large as practical considerations will permit, in order that there may be ample opening at short cut off to avoid undue wire drawing.

Variable Load.—In nearly all conditions of service the load varies more or less, and the nature of this variation and its extent will have an important bearing in determining the type of engine to be used and degree of economy to be expected.

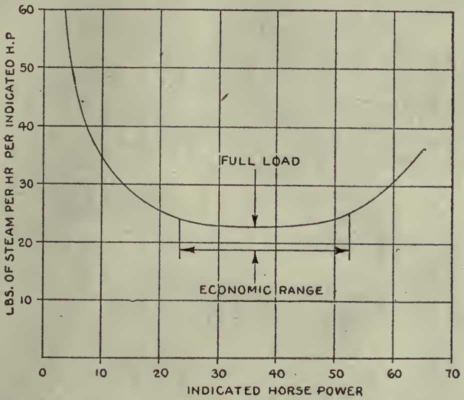


FIG. 3,177.—Characteristic load curve illustrating relation between economy and variable load.

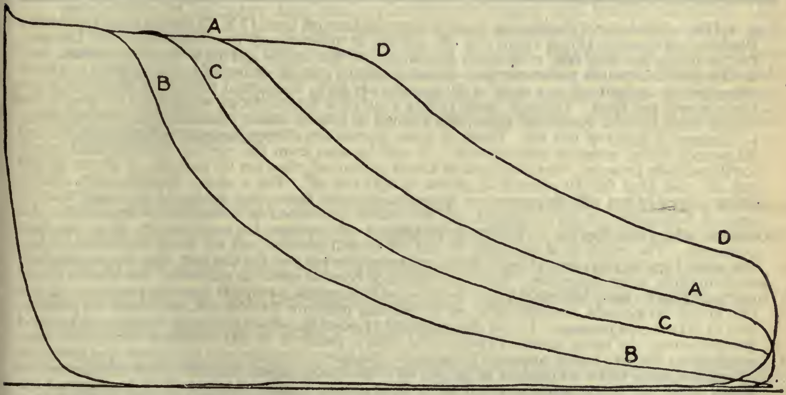


FIG. 3,178.—Diagram illustrating Buckeye power ratings, based on economical range of power, advisable under 100 pounds boiler pressure. This range, as represented by the diagram, is bounded by the lines AA, and BB, the first representing 60 pounds *m.e.p.*, and the latter 40; the "rated power" is based on the mean of the two, or 50 pounds, represented by line CC. The "ultimate capacity" is based on 70 pounds *m.e.p.*, represented by line DD, which, although not the utmost power the engine is capable of exerting, is about the greatest under which good regulation can be expected, unless a higher boiler pressure be carried, or the adjustments have been made with reference to such an excessive duty.

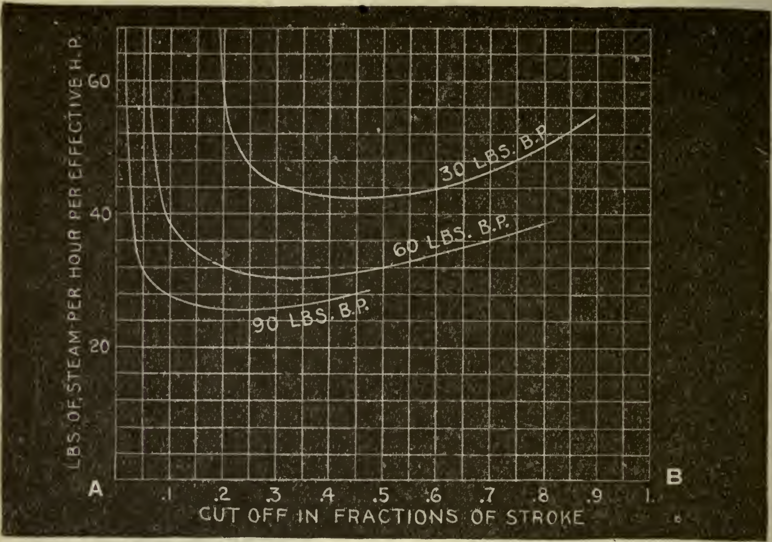


FIG. 3,179.—Load curve illustrating most economical cut off for a 17 × 30 simple non-condensing Buckeye automatic riding valve cut off engine, as determined from tests by Prof. Denton. In the figure the line **AB**, represents the stroke of the engine. This is equally divided into tenths and half-tenths to denote the various points of cut off, or periods of admission. Above these points are laid off to a scale of 20 pounds per inch, the actual pounds of water used per horse power per hour. Through these points a curve is drawn, from which may be seen more clearly than from a numerical table, the change in rate of water consumption corresponding to changes in point of cut off. Three of these curves are shown, corresponding to 90, 60 and 30 pounds boiler pressure respectively. It will be seen from the curves, that the best result for 90 pounds pressure, was obtained at about ¼ cut off; that for 60 pounds, at about 33/100 cut off, and that for 30 pounds at about 45/100 cut off. For a short distance each side of these points of cut off, the economy is practically the same, but it should be noted that in case of very early or very late cut off, the water consumed is considerably increased. An engine, therefore, having a widely fluctuating load, cannot be expected to give the best economy. The relation of economy to cut off is well pointed out by these experiments, but the actual water rates would have been somewhat better but for the fact that the engine had uncommonly large clearance. Had the clearance been as small as might be, the results would probably have been about 3½ per cent. more favorable, or at 90 pounds pressure, ¼ cut off, about 26 pounds consumption; at 60 pounds pressure 35/100 cut off, about 30 pounds, and at 30 pounds pressure, ½ cut off, about 41½ pounds, which amounts correspond to good performance. In some similar experiments with a small (7 × 14) Buckeye Automatic non-condensing engine, boiler pressure 75 pounds, the same experimenter has found the best rate of consumption to be 30 pounds at ¼ cut off. It would appear here that the most favorable point of cut off is the same in small as in large engines, while the matter of economical use of steam is in favor of the larger engine and high pressure. It should be borne in mind that these results are deduced from refined tests, extending over a considerable period, the engines being in perfect condition as to leakage, etc., and the steam collected, condensed and carefully weighed after the work was performed, thus eliminating errors as far as possible. The steam was also practically free from moisture. In the tests of the above figure it contained less than one per cent. From these and other experiments an approximate rule may be stated that the best cut off, expressed in fractions of the stroke, may usually be determined by dividing 100 by 42 times the square root of the boiler pressure—this pressure being reckoned from a

For every engine *there is a certain load, and certain set of working conditions, such as initial pressure, cut off, speed, etc., at which it will work with maximum economy. This is known as its* ***"full load."***

The load may be varied a reasonable amount either above or below the full load without perceptibly impairing the economy. This load range is called the *economic range*, and evidently the economic range is different for different types of engines.

The economic range is greater for a simple engine than for a compound or triple engine.

If the load be varied beyond the limits of the economic range, there will result a marked decrease in economy, the relation between variable load and economy being shown by a load curve as in fig. 3,179.

In considering any particular engine for a given set of working conditions, a study of curves, such as shown in the figure plotted for the engine in question, will aid in determining if the engine be suited to the service.

Fig. 3,181 shows a similar curve based on steam per horse power hour and variable cut off.

Very often the conditions are such, as for instance in a new and growing plant, as to require an engine too large for the service, the result being that the engine will work with poor economy until the growth of the plant increases the load to the economical range of the engine. There are several solutions to a problem of this kind: 1, a simple engine may be installed, proportioned to carry the load at moderate speed and boiler pressure, later a smaller pulley may be used and the speed increased, also the boiler pressure, thus adapting the engine to quite a large increase in load without material change in economy; 2, a simple engine may be installed and later a low pressure cylinder added converting the engine into a compound, giving considerable increase in power and economy without adding additional boiler capacity.

The performance of multi-stage expansion engines with a variable load is a complex matter and the effects of the various methods of governing on the steam distribution as explained in the chapter on Multi-Stage Engines (pages 603 to 642) should be studied and thoroughly understood.

Although simple engines seldom give trouble when running on light loads the same cannot be said of multi-cylinder engines. Often the power is very unevenly divided between the cylinders, especially in triple engines.

A compound or triple engine running at very light load is subject not only to loss of economy and to irregular action, but the inside surface of the low pressure cylinder is liable to be cut or abraded.

5. Methods of Governing

Throttling.—In general, a throttling governor may be considered *as not much more than a* ***makeshift*** *so far as economy is concerned.*

Under ordinary conditions this form is very wasteful and is accordingly confined to the cheaper and less efficient types of engine, or where the cost of fuel is very low. However, if the load vary greatly, throttling would be more efficient than variable cut off near the limits of the load range,

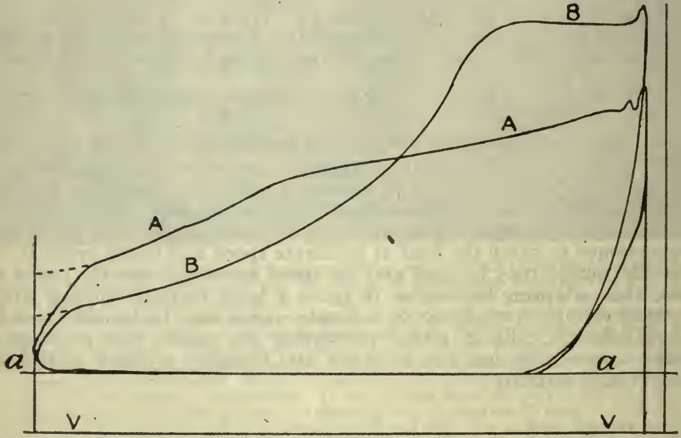


FIG. 3,180.—Diagram illustrating Buckeye comparison of ***automatic cut off*** with ***throttling*** regulation. The diagram is reproduced from actual card taken from Buckeye engines. Both represent very favorable loads and each shows good results for its type of engine. The throttling card AA, develops 40.57 pounds *m.e.p.* with 37 lbs. absolute terminal pressure, while the cut off card BB, develops 41 pounds *m.e.p.* with only 27 lbs. absolute terminal pressure. Thus the cut off engine was developing 41 pounds of work with an expenditure of its cylinder full of steam at 27 pounds pressure, while the throttling engine developed but 40½ pounds of work with cylinder full of steam at 37 pounds pressure per square inch. For the purpose of comparison it will be sufficiently near correct to assume both diagrams to have same *m.e.p.* By the usual methods of calculating theoretical water consumption it will be found that the rates are (omitting nearly equal fractions) 31 and 23 pounds per *i.h.p.* per hour, for the diagrams A, and B, respectively, showing a saving of 25.8 per cent., by the use of automatic cut off over throttling regulation, but since there will be a trifle less internal condensation involved in the throttling process, owing to the lower initial pressure and temperature, this saving will not quite hold in practice against as near perfect throttling performances as is here represented. Perhaps about 22 or 23 per cent. would be more nearly the actual saving. On the other hand, such throttling performance is very rarely realized in general practice, and it is a very common experience to effect saving by such substitutions of 30, 40 or even 50 per cent.

especially if the fixed cut off of the throttling engine occur early in the stroke, even so, the advantage of throttling over variable cut off in such case would depend on whether the load was near normal or near the limits the greater part of the time.

For loads of such nature the best economy would be obtained by a combination of throttling and variable cut off, the governing mechanism being so arranged that the throttling action would prevent the cut off varying beyond economic limits.

Variable Cut Off.—This is, in general, *the* ***accepted method*** *of engine control where economy is sought.*

In considering variable cut-off, the nature of the steam distribution corresponding to the different methods employed should be understood.

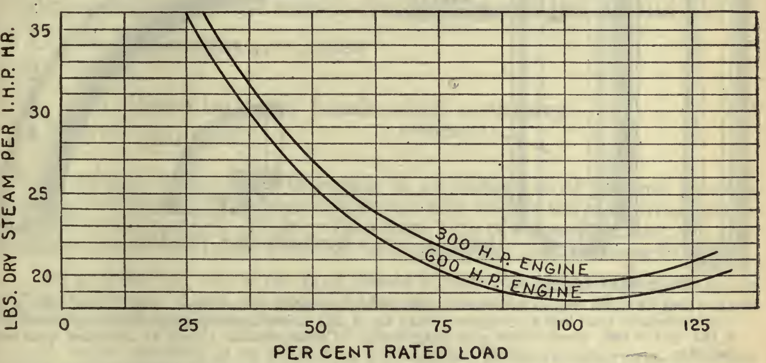


FIG. 3,181.—Economy curves for various sizes of compound non-condensing four valve engine variable load. Boiler pressure 150 pounds.

The cut off may be varied:

1. By shifting eccentric.
2. By double eccentric (independent cut off valve).
3. By the so called expansion valve gears as the link motion, radial motion, etc.

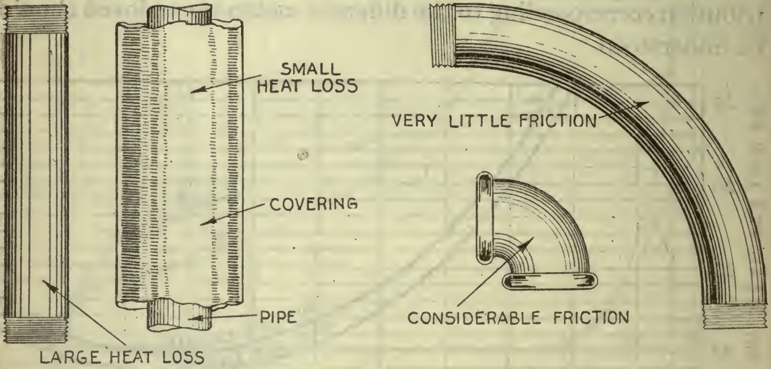
The points to be considered are:

1. Where both admission and exhaust are controlled by one valve, as by the method of ***combined*** *variable angular advance and variable throw* (shifting eccentric) early cut off is obtained at the expense of too much

compression and not enough port opening for either steam or exhaust, which more or less offsets the economy that would otherwise be due to the short cut off.

2. Because of the defects mentioned in 1, separate and independently driven exhaust valves are provided in engines designed for greater efficiency. Thus the cut off may be varied any amount without disturbing the exhaust and compression.

3. In certain types of gear such as the riding cut off, the clearance is considerable, and, where the economy due to a high number of expansions in a single cylinder is desired, the effect of clearance should be carefully considered, and in this connection it should be understood that it is not the percentage of clearance referred to the cylinder volume, but the percentage referred to the cylinder volume up to cut off that affects economy.



FIGS. 3,182 to 3,185.—Bad and good construction for main steam pipe. Fig. 3,182, bare pipe —considerable heat lost as compared with fig. 3,183, showing properly insulated pipe. Figs. 3,184 and 3,185, close elbow and easy bend. ***Close elbows should be avoided wherever possible.***

In determining what governor is best suited for any given conditions of operation, not only must economy be considered, but the nature of the requirements as to regulation (speed variation), and a governor must be employed which will maintain the speed as near constant as is necessary. For instance, in electric lighting a practically constant speed is essential but for a traction engine no such refinement is necessary.

6. Auxiliary Apparatus

Main Steam Pipe,—An inspection of almost any steam plant (especially those of moderate or small size) will show that little or no provision has been made for economy.

Long, leaky, and bare pipes are to be seen, containing an unnecessary number of elbows but no expansion joint.

Steam begins to lose its heat ***before*** *it gets out of the boiler.*

What then must happen when it passes through such a pipe line as above described?

Not only is heat lost by radiation and leakage, but there is a loss of pressure due to frictional resistance of the long and crooked pipe line. Sometimes there is a difference of 5 or 10 pounds in pressure between the boiler and engine, but in well designed plants it should never be greater than 2 lbs.

Accordingly for best results the main steam line should be:

1. Of proper size.
2. Short.
3. No elbows but easy bends when necessary.
4. Well insulated.

If the pipe be too small there will be an undue loss of pressure between boiler and engine; if too large, the steam flow will be too slow, resulting in loss by condensation. There is then some rate of steam flow that will give the *minimum* loss in the pipe due to condensation and pressure reduction.

In general *the size of the pipe should be such that the velocity of the steam flowing through it will not exceed 8,000 feet per minute, or 10,000 feet in the case of very large plants.** ϕ

A better method is to assume a permissible pressure drop and calculate the pipe by aid of formulae and tables. Since the loss of pressure is proportional to the length of the pipe, it should be as short as possible, that is, the engine should not be placed at an unnecessary distance from the boiler. Very few consider the loss of pressure due to an elbow.

In calculating this loss, an elbow may be taken equivalent to a straight length of pipe 40 diameters long. By using an easy bend both the length of pipe and loss due to an elbow is reduced.

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*NOTE.—According to Fernal and Orrok, 8,000 feet was considered maximum in the days when 125 pounds steam pressure was carried without superheat. With 200 pounds and superheat up to 200° Fahr. the minimum steam speeds are much higher, and very few engineers are using a minimum less than 8,000 feet per minute—the minimum in some cases running as high as 18,000 feet with no bad results.

ϕNOTE.—8,000, and 10,000 ft. per minute are proper for $^6/_{10}$ cut off, but the shorter the cut off the greater the permissible speed. See chapter 52 on Engine Design and chapter IV. of Prof. Fessenden's excellent book on "Valve Gears."

Reducing Valves.—The practice of reducing the pressure between the boiler and engine is sometimes resorted to.

1. To keep the pressure constant at the engine, and,
2. To dry or slightly superheat the steam.

Constant pressure is the chief object sought, because very little superheat can be obtained by reducing the pressure, and even then, the steam is more economically superheated in the boiler by absorbing heat which otherwise would pass up the chimney.

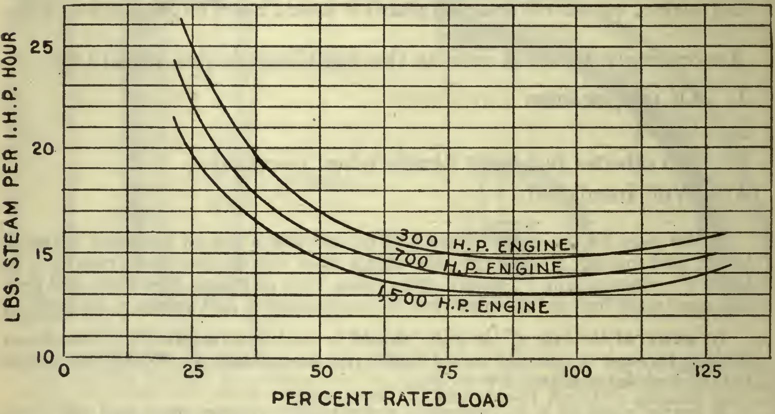


FIG. 3,186.—Economy curves for various sizes of Corliss compound condensing engine, variable load. Boiler pressure, 160 pounds; 26 inch vacuum.

If a reducing valve be used and the pressure be reduced from say 300 to 250 pounds, it would (assuming the steam reached the valve without moisture), be superheated only about 10 degrees, too small an amount to appreciably increase the economy.

Moreover, boilers in general do not furnish dry steam and with the usual percentage of moisture, dry steam at most would be all that could be expected at the engine by reducing the pressure.

In general, it would be preferable to install automatic dampers, and seek to improve the firing methods rather than resort to reducing valves.

Separators.—According to tests made by Prof. Carpenter, the moisture in steam leaving a Stratton separator was less than 1 per cent when that in the steam supplied ranged from 6 to 21 per cent. Although ordinary boilers not having superheaters furnish steam containing very little moisture, there is more or less condensation in the pipe between boiler and engine, hence a separator should be used to trap this condensate and suitable means provided for returning it to the boiler so that the heat which it contains may be saved.

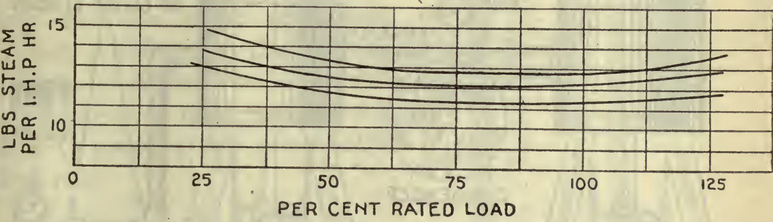


FIG. 3,187.—Economy curves for various sizes of grid iron valve compound condensing engines variable load with 100° Fahr. superheat. Boiler pressure, 165 pounds.

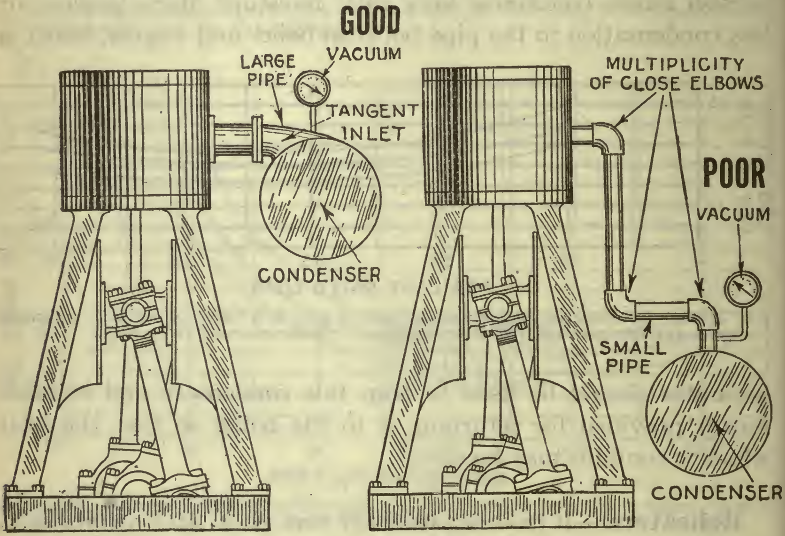
Reheaters.—It has been stated (*Trans. A.S. M.E.*, vol. XXV, 482-492) that the ***reheater*** *does for the low pressure cylinder of a multi-stage expansion engine just what superheat does for a single cylinder.*

This is a very clear way of expressing it, and as the object of a reheater is so simple no further explanation should be necessary.

According to Marks, adequate reheating involves superheating the steam in the receivers from 30° to 100° Fahr. With the latter amount, Marks has shown that the efficiency of the compound engine may be increased 6 to 8 per cent. Good reheating makes low pressure jackets unnecessary.

As a rule, not nearly enough reheating surface is installed. With moderately superheated steam in both cylinders, the steam consumption is about constant for the range from ½ load to 1¼ load.

Exhaust Pipe.—Perhaps less consideration is given to this item than to any other part of a power plant, accordingly most engines are operating with more or less unnecessary back pressure which represents a constant loss and which could have been avoided at nominal expense by intelligent designing. The subject has been fully explained under the heading Back Pressure (page 1,650), and it is unnecessary to add anything more, than briefly state the essential requirements for obtaining the best economic results. These are:



FIGS. 3,188 and 3,189.—The *right* and the *wrong* way to pipe engine to condenser.

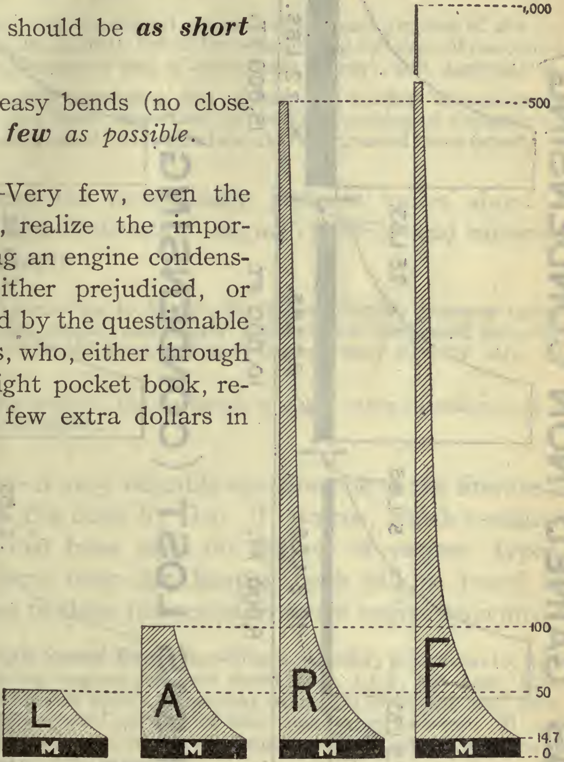
1. The exhaust pipe should be of such size that the velocity

of the exhaust shall not exceed 4,000 feet per minute; on long lines a somewhat larger pipe should be used.

2. The pipe should be ***as short*** *as possible.*

3. Use only easy bends (no close elbows) and *as* ***few*** *as possible.*

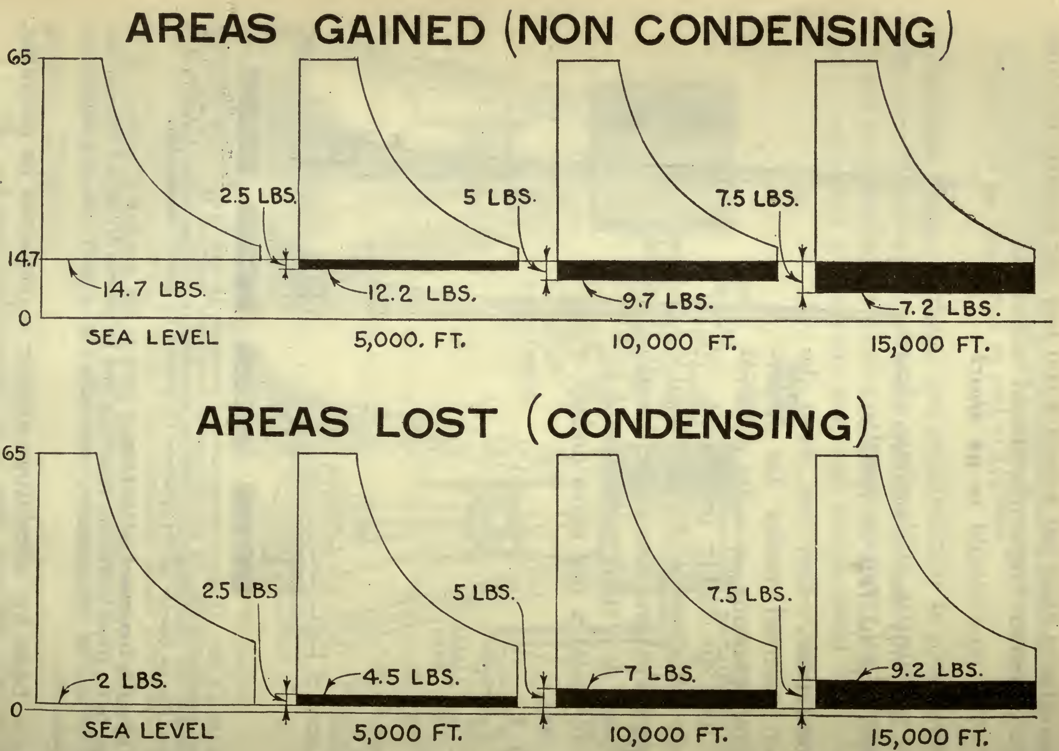
Condenser.—Very few, even the better informed, realize the importance of operating an engine condensing; they are either prejudiced, or unduly influenced by the questionable practice of others, who, either through ignorance or a tight pocket book, refuse to spend a few extra dollars in first cost.



FIGS. 3,190 to 3,193.—Diagrams showing effect of pressure or the gain by condensing, expanding to atmospheric pressure. In each diagram the equal solid black area **M**, is that due to the condenser. By comparing this with the progressively increasing areas, **L,A,R,F**, it is evident that ***the less the initial pressure,*** the ***greater the gain by condensing.***

Usually, ***too much attention*** *is paid to the practice of others.* It is well to do more individual reasoning and in general to regard the practice of others as questionable, especially in regard to non-condensing operation.

As a basis for calculating whether a condenser be desirable,



FIGS. 3,194 to 3,201.—Theoretical diagrams showing effect of altitude on engine capacity. Figs. 3,194 to 3,197, non-condensing; figs. 3,198 to 3,201, condensing. The diagrams show that as the elevation increases, the capacity ***increases*** *for* ***non-condensing*** *operation* and ***decreases*** *for* ***condensing*** *operation.*

it should be understood that the *minimum* saving in fuel is 15 per cent, better results being obtained according to conditions.

The chief items to be considered are cost of fuel and expense of the circulating water and its quality, the latter determining the type of condenser to be used, also difference in cost of attendance (if any), and depreciation.

With the constantly increasing cost of fuel, the subject of condensers is becoming more and more important, hence the problem of determining whether a condenser should be installed should be entrusted to an expert.

Altitude.—Since the atmospheric pressure varies about 1 pound per 2,000 feet of elevation, engine efficiency and capacity is affected accordingly.

In this elevation, according to Marks, the mean effective pressure varies about ¾ pound for simple, and nearly 1 pound for compound non-condensing engines. The efficiencies of such engines vary directly with the altitude.

With condensing engines, the variation is in the other direction, but is so slight as to be negligible.

Engine Tests.—A very valuable contribution to the literature on this subject is the book by Geo. H. Barrus, which contains the results obtained from tests on engines of various types. The following items from Mr. Barrus' book will be found of considerable value to those interested in steam engine economy.*

Automatic High Speed Engines.—Single cylinder, piston vavle, high speed non-condensing engines will now develop an *i.h.p. per hour*, from 25 to 35 lbs. of saturated steam, depending upon size of engine and pressure available. Compound engines of this type range between 20 and 25 lbs., non-condensing, and when condensing with 26″ effective vacuum, from 17 to 23 lbs. may be expected.

Four Valve Engines.—Single cylinder, non-condensing, medium speed, four valve engines will range in economy between 21 and 26 lbs. of dry saturated steam, condensing between 17 and 22 lbs. *per i.h.p. per hour*. Compound, non-condensing engines of this type should be expected to operate upon from 16 to 21 lbs., and when condensing with 26″ effective vacuum, from 13 to 17 lbs.

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*NOTE.—The author heartily recommends Mr. Barrus' book on engine tests as it is the work of a noted authority on the subject; he has also a similar book on boiler tests.

"Gridiron" Valve Engines.—Under condensing conditions and good steam pressure, the steam consumption *per i.h.p. per hour*, varies between 14 lbs. for units of 500 *kw.* capacity to about 12 lbs. in units of 2,000 *kw.* and above. Under superheated conditions, economy has been obtained upon 5,000 *kw.* units of this type better than 11 lbs. of steam *per i.h.p. per hour*. The mechanical efficiency of these larger engines is over 96%. The horizontal vertical cross compound arrangement, in which friction in the engine itself is reduced to a very small amount, owing to the arrangement of cylinders and the resulting uniformity in the crank effort has been shown a combined mechanical and electrical loss in both direct connected units, as little as 6%, which means a net efficiency of both engine and generator together of 94%.

Steam Turbines.—In order to operate economically the most improved type of steam turbine, it is imperative that there be maintained:

1. High steam pressure.
2. Steam of a high degree of superheat.
3. Highest possible vacuum.

The last is the most important of the three things. Under these conditions excellent results have been obtained under a steady uniform load. It is, however, under a very variable load factor where extremely small fractional loads must be met, that the steam turbine is distanced by its older competitor. This is diametrically opposite to the original claims made for the turbine, but if the reports of the tests conducted by admittedly disinterested and competent engineers are to be regarded, the small amount of available data of station tests is consulted, and, above all, the guarantees made by the turbine manufacturers under competition for prospective plants are to be viewed as a criterion, claims for superior fractional load economies are entirely without basis.

Effect of Superheat on Cylinder Condensation

No. of test	Degrees Fahr. of superheat	Cut off	Proportion of cylinder condensation and leakage	Cylinder condensation and leakage derived from curve for ordinary steam	Proportion of condensation reduced by superheat
1C.	82	.392	.947	.053	.114
4.	25	.233	.766	.234	.021
8A.	37	.247	.819	.181	.062
8B.	37	.165	.747	.253	.081
9A.	24	.185	.820	.180	.127
9B.	24	.225	.836	.164	.097
15.	59	.281	.895	.105	.110
Average	41				.087

CHAPTER 52

HOW TO DESIGN AN ENGINE

It might be inferred from the title of this chapter that the subject is to be treated in a popular manner—not exactly popular, but with common sense and in a rather unconventional style. In fact it may be regarded as a ***protest*** *against the tendency of engineers to be influenced by what may be termed the* ***power of suggestion,*** that is, the tendency to blindly follow the practice of others.

Such men as Watt, Corliss, White, Stumpf, Diesel, etc., do not belong to this class but may be called *independent thinkers;* men who are not satisfied with mediocre attainments in engineering, but who seek something better. Some engineers accept standard practice, in certain instances that they know to be faulty because they lack the strength of their convictions, being afraid to introduce something new lest it do not come up to their expectations.

For instance, in small and medium size city water supply pumping plants, wasteful direct connected pumps are in most cases installed, although very little reasoning will show that the conditions at the water end and at the steam end of the pump are directly opposed so that the two ends should be independent to obtain high economy, as in the arrangement shown in fig. 3,202. Here large cylinders and low piston speed are eliminated and a small cylinder and high piston speed substituted with belt drive. However, this is *different* from the usual practice—the belt might slip, or break, or run off; in fact, a multitude of imaginary catastrophes might happen, and so the old direct connected pumps are installed notwithstanding the large coal bills that accompany them. Instances such as this represent what might be termed *engineering crimes*, and there are many.

Again, there are engineers who have not the inventive instinct and rely on the practice of others. It is very well to listen to what the other fellow has to say, whether he be a professor or a helper in a garage, but *do not accept his statements as correct* without satisfying yourself as to the facts by *independent reasoning* or *investigation*.

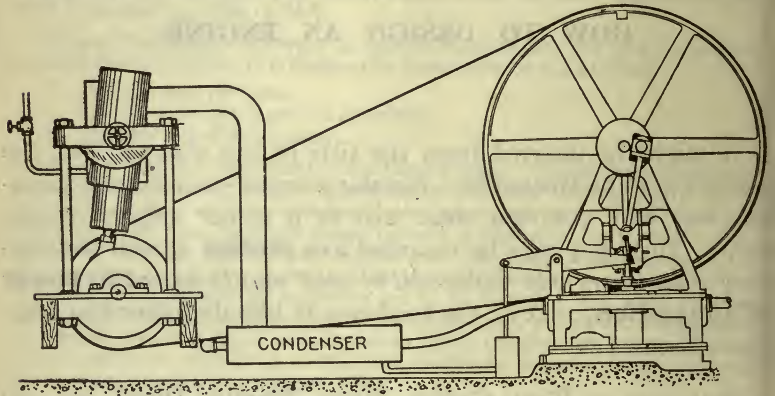

FIG. 3,202.—Graham jacketted, transfer expansion oscillating engine, and Dunham power pump with belt transmission. ***It must be evident*** that this arrangement permits the engine and pump to work under the most favorable conditions for economy. Hence by proper proportion of pulleys, the engine may run at ***high speed,*** thus reducing the size of the cylinder and loss by condensation leakage, etc., and the pump may be run at ***slow speed,*** thus reducing the loss by slip, and eliminating water hammer, slamming of valves, etc.

*The chief aim of this chapter is to appeal to the engineer to

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*NOTE.—Not only in engineering but in other sciences and arts are we unduly influenced by the opinions of others. For instance, in music, there are still some singers, a few musicians (?), and other nondescripts who never miss a performance of Handel's Messiah, yet who *could not be hired* to attend a performance of Verdi's Requiem or Grieg's piano concerto, Bruno Oscar Klein's Messe Solennelle op. 82, or Tschaikowsky's Symphonie Pathetique,—or listen to Arthur Depew's wonderful improvisations on the organ. ***Why?*** Their grandmothers, great-aunts and some early singing teachers of the *bel canto* and *coloratura* species never missed a performance of the Messiah and thought it was *great stuff!* Thus influenced by the suggestion of others they accordingly hold the same opinion, not having sufficient intelligence to use their common sense in the matter. Handel and some of the other so-called great composers were great carpenters but no architects. *How could they be without* ***modern harmony?*** Why be satisfied with an old spinet or clavichord when a modern Steinway grand is infinitely better? Why prefer an antiquated "horseless carriage" to a Stutz, Packard, or Rolls Royce? Why listen only to primitive music and object to old style plumbing when the modern music (but not the super-modern as has been inflicted on us by some French futurists) is over a hundred million times better? ***Why?***

think for himself and use common sense, rather than follow blindly the practice of others.

The particular engine selected to illustrate how to design an engine is an original idea of the author intended to replace a standard though very uneconomical type.

It is to be expected that the author's engine will be criticised because it is something new, different from the usual practice and might "slip a cog," or something else equally dreadful might happen.

How to design an engine is illustrated by the design of an engine to meet the requirements given in the example which follows, this being presented in the main text, and some of the mass of rules formulæ, etc., of the leading authorities, given independently in small type as foot notes. Some of these rules would be better printed in still smaller type or no type at all, representing as they do empirical formulæ for "rules of thumb," these being based on practice instead of on the strength of materials, should be regarded with suspicion and avoided as much as possible.

Example.—Design a single cylinder marine engine for a moderate speed boat, to develop 30 indicated horse power at full load.*

There are some people who think of nothing but first cost. They select the cheapest article regardless of future expense. A large percentage of this class consists of steam boat owners, as is indicated by the labored puffings of ***non**-condensing* lighters and tugs whose engines are too small for the job, and, operated without condensers, are very wasteful. However, as mentioned, first cost is the chief item with their owners, hence the selection of these extremely robust "puffing Billies" or steam eaters. To illustrate, a prominent concern once built a 22×26 lighter engine which was rated at 500 horse power

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*NOTE.—The author originally intended to design a 500 horse power engine for a steam lighter, but selected the small size with the idea of building one at some future time to experiment on a small scale with the steam jacket. The small engine will, however, serve just as well to illustrate the principles of engine design.

At ordinary speed, say 125 revolutions per minute (*r.p.m.*), this engine would require a mean effective pressure (*m.e.p.*) of 79.4 lbs. per sq. in., and with the usual late cut off, correspondingly high initial and terminal pressures resulting in poor economy as indicated by the card, fig. 3.203, This means large boilers, large coal bunkers, and large condenser (if one be used), all of which occupies valuable space which otherwise could be used for cargo. Moreover the excess weight of the plant increases the draught which in turn reduces the speed; thus more coal is required on this account and also on account of poor economy.

Watt has said: *the supreme excellency in machinery is in its simplicity.*

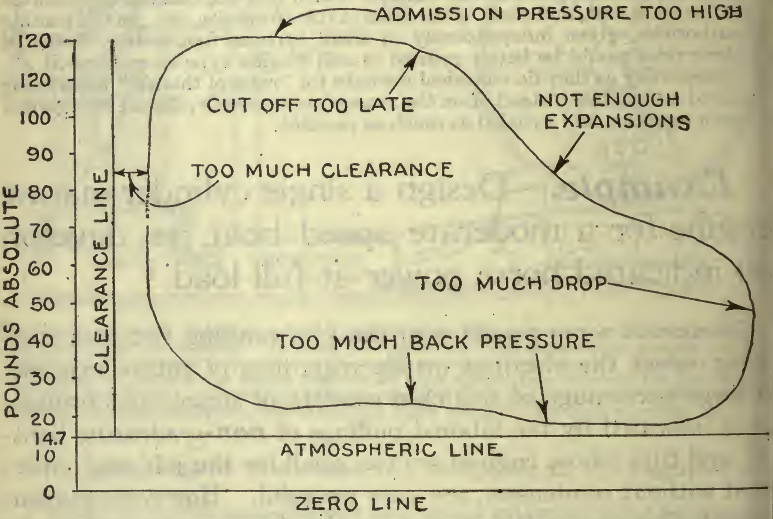


FIG. 3,203.—Card from an ordinary marine engine of the type usually installed in steam lighters, having a riding cut off with resultant large clearance, poor steam distribution, etc. The diagram shows six of the numerous reasons for poor steam economy.

Accordingly a single cylinder engine would be very excellent if it were as economical as a compound, and the author has this idea in view in designing the engine illustrated in this chapter.

The ordinary compound (without independent cut off) is usually designe to cut off at $^6/_{10}$ stroke in the high pressure cylinder which, with a cylinde ratio of 4:1 (the ratio usually but not always judiciously employed), would give

$$(1 \div {}^{6}/_{10}) \times 4 = 6.64 \text{ expansion.}$$

In this type of engine the various losses, such as condensation, wire drawing, drop or free expansion in the receiver will result in a diagram factor of about .95 and a water consumption of about 20 lbs. The water consumption of the 22×26 working under the conditions of fig. 3,203, would be about 40 lbs. or twice that of the compound.

The Theoretical Diagram.—To compete with the performance of the compound, the following operating conditions are selected as being the most desirable for economy in a single jacketted cylinder using saturated steam: Initial pressure 80 lbs. (gauge); 7 expansions or as near that as the valve gear will permit; 28 inch vacuum (.943 lbs. say 1 lb. absolutely) back pressure.

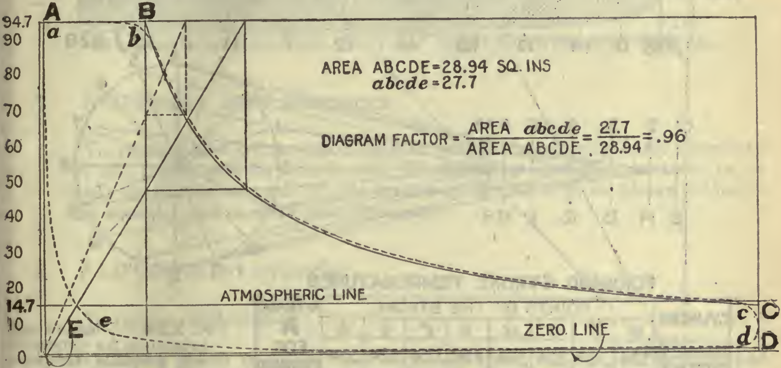
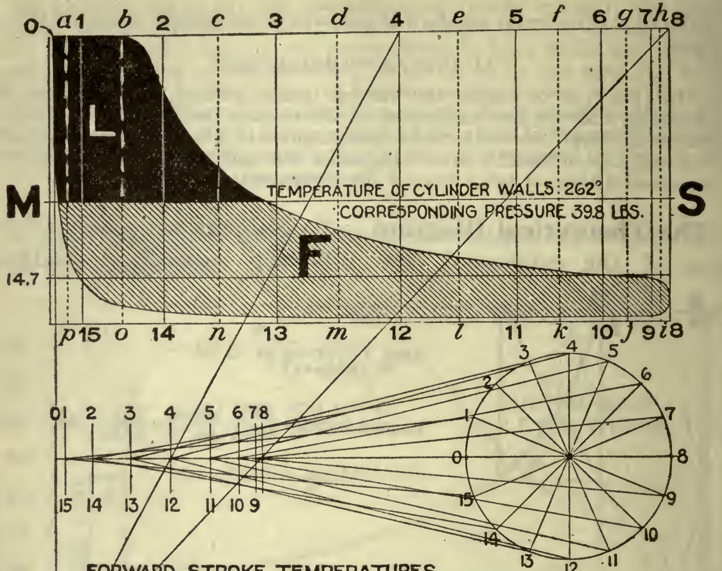


FIG. 3,204.—Theoretical diagram and expected card superposed to determine the diagram factor.

In fig. 3,204 is a theoretical diagram, A,B,C,D,E, corresponding to these conditions and the probable temperature of the cylinder walls is obtained as explained in fig. 3,205, in order to aid in sketching in the expected card with respect to the effect of condensation and re-evaporation upon the card.

The Expected Card.—To obtain compound economy in a



FORWARD STROKE TEMPERATURES

CYLINDER	POINTS OF THE STROKE								AVERAGE
	a	*b*	*c*	*d*	*e*	*f*	*g*	*h*	**A**
JACKET SURFACE	324	324	324	324	324	324	324	324	FOR FORWARD STROKE
INTERIOR SURFACE	324	324	280	250	231	217	213	210	
AVERAGE	324	324	302	287	278	271	269	267	290

RETURN STROKE TEMPERATURES

CYLINDER	POINTS OF THE STROKE								AVERAGE
	i	*j*	*k*	*l*	*m*	*n*	*o*	*p*	**B**
JACKET SURFACE	324	324	324	324	324	324	324	324	FOR RETURN STROKE
INTERIOR SURFACE	134	114	114	114	114	136	170	238	
AVERAGE	229	219	219	219	219	230	247	281	233

TEMPERATURE OF THE METAL $(A+B)\div 2 = \frac{290+233}{2} = 262°$

Fig. 3,205.—Diagram and tabulation for determining probable temperatures of the cylinder walls. A number of positions, *a*, *b*, *c*, etc., are found corresponding to equal angular movements of the crank and the temperatures of the steam in the cylinder at their points determined from the steam table. These values are subtracted from the constant temperature of the jacket steam and average values **A** and **B**, calculated for the forward and return stroke. The average of these two average values gives the average temperature for the two strokes, which approximates the temperature of the metal indicated by the line **M S**. Accordingly some *condensation* may be expected to occur during the portion **L** of the card in which the steam is at a temperature *higher* than that of the cylinder walls and *re-evaporation* during the portion **F**, in which the steam is at a temperature *lower* than that of the cylinder walls. ***The ideal condition*** sought by *jacketting* is to reduce this condensation and re-evaporation to a minimum, exhausting the steam in its saturated state.

single cylinder engine, the design should be such as to give a very high *diagram factor* so that, as nearly as possible, the full benefit due to the ratio of expansion and temperature range will be obtained. Accordingly the design should embody the following features:

1. Quick acting valve gear.

   To reduce wire drawing.

2. Separate admission and exhaust valves.

   To secure a semi-uniflow effect and to give independent exhaust.

3. Practically zero clearance.

   To permit late compression and to avoid abnormally early apparent cut off, for as will be found with the ***shifting*** *eccentive* (the type to be used) *the* ***earlier*** *the cut off, the* ***more difficult*** *it is to obtain adequate port opening and to avoid wire drawing.*

4. *Full* jacketted cylinder.

   To reduce condensation.

With these items in mind the expected card, *a,b,c,d,e,f,* is sketched within the theoretical diagram as shown in fig. 3,204, from which the diagram factor is obtained. Although a small error in the diagram factor is not of vital importance it is desirable that the engine performance at full load closely approximate the expected card to obtain maximum economy.

Expected M.E.P.—The theoretical mean effective pressure corresponding to the given conditions of operation as found in fig. 3,204 is 38.88 lbs. and the diagram factor .96, hence the expected mean effective pressure upon which the size of cylinder is based is

$$38.88 \times .96 = 37.34, \text{ say } 37 \text{ lbs.}$$

NOTE.—The diagram factor according to Seaton, varies (for various types of engines) from .6 for high speed war ship engines to .94 for expansive engine, special valve gear, or with a separate cut off valve, cylinder jacketted.

Revolutions per Minute (*r.p.m.*).—For every size and type of boat there is a certain propeller speed which gives the greatest efficiency, that is, which causes it to operate with the least *slip** and consequent loss of power.

Size of Cylinder.—The preceding paragraphs indicate that a *large* cylinder will be necessary to develop the given power. The cylinder is *not* large but only seems so when compared with the size usually selected for the same power because the latter, as before stated, is too small for the job, the object being simply *to get as much power as possible per lb. of metal and per cu. ft. of space occupied by the engine regardless of economy.*

A little consideration will show that not only is *the weight and space thus saved is transferred to the boiler*, auxiliaries, and additional coal supply necessary, but the weight and space occupied by the complete power plant is greatly increased with heavier operating expenses. Accordingly the mental process of the designers who produce such faulty power plants is is not clear to the author.

In determining the cylinder dimensions, to begin with, the experienced designer knows from his sense of proportion, approximately the size of cylinder that will develop a given power.

From this sense of proportion, a range of strokes, or diameters may be selected in preparing a table of various combinations of strokes and diameters that will develop the given power, it being advisable to study such a table to arrive at the most desirable dimensions.

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*NOTE.—There are two kind of slip: *real* and *apparent*. The ***real slip*** is the velocity (relative to water at rest) of the water projected sternward by the propeller; the ***apparent slip*** is the difference between the speed of the ship and the speed of the screw, that is, the product of the *pitch* of the screw by the number of revolutions. The term "pitch" is analogous to the term "pitch of the thread" of an ordinary single threaded screw. Thus, if there were no slip, a propeller would push the boat forward in one revolution a distance equal to the pitch of the propeller.

NOTE.—***Steam Consumption of engine at various speeds.*** (Profs. Denton and Jacobus, *Trans. A. S M. E.*, x 722. Tests on a 17×30 engine, non-condensing, fixed cut off, Meyer valve.) ***Conclusions:*** There is a distinct gain in economy of steam as the speed increases for ½, ⅛, and ¼ cut off at 90 lbs. pressure. The loss in economy for about ¼ cut off is at the rate of 1/12 lb. of water per *h.p.* for each decrease of a *r.p.m.* from 86 to 26 rev., and at the rate of ⅝ lb. of water below 26 revs.

To facilitate calculations of the table, use is made of the *horse power constant* or product of the known factors; these are

1. Formula coefficient .000004
2. *M.e.p.* 37 lbs.
3. *R.p.m.* 250
4. Horse power 30.

Now the horse power formula reduced to its lowest terms (as explained in Guide No. 1, page 82), is

$$\mathbf{H.P. = .000004D^2LNP} \quad \ldots\ldots (1)$$

and substituting the values just given

$$30 = .000004 \times D^2 \times L \times 250 \times 37 \quad \ldots\ldots (2)$$

Solving the equation (2) for L (rather than for D, to avoid extracting the square root)

$$L = \frac{30}{.000004 \times D^2 \times 250 \times 37} = \frac{810.81}{D^2} \quad \ldots\ldots (3)$$

Now for cylinder diameters

8 8½ 9 9½ 10

inches, the corresponding lengths of stroke as found by substituting these values in (3), and calculating by slide rule, are

12.7 11.2 10 9 8.1

inches, and arranged as a table.

Cylinder Dimensions

(For 30 horse power at 250 *r.p.m.* and 37 lbs. *m.e.p.*)

Diameter..................	8	8½	9	9½	10
Stroke....................	12.7	11.2	10	9	8.1

It should be remembered that the chief aim in the design of this engine is economy—*compound economy with a single cylinder*, hence the question: Is jacketting more efficient with a short or long stroke engine?

The long stroke engine is the more efficient (as explained in figs. 3,166 and 3,167), because *the ratio of the jacketted surfaces to the cylinder volume* *is greater*. Proportioning the stroke on this basis when carried to extreme conflicts with the marine condition which requires a low center of gravity, hence a compromise must be reached, and inspecting the table the dimensions $8\frac{1}{2} \times 11.2$ may be selected as most desirable proportions, all things considered.

The steam engine (unlike the gas engine) has a considerable overload power range, this range being very great in the type of engine here considered, hence no such refinement as .2 in. is necessary in fixing the length of stroke which therefore may be taken as 11 ins., making the size of cylinder as selected

8½ X 11[*]

At this point it is well to check the slide rule calculation and determine the actual power of the selected size, thus

$$\text{Horse power} = .000004 \times (8\tfrac{1}{2})^2 \times 11 \times 250 \times 37 = 29.4$$

Although this is .6 horse power short of the required power, the deficiency is of no consequence because: 1, a very slight lengthening of the cut off will give full power, and 2, it is somewhat doubtful whether the valve gear will permit using as short a cut off as $^1/_7$ with proper steam flow.

Cylinder.—Since the valves are to be placed in the heads, the cylinder of this engine will be a very simple casting. Th size is small enough to permit casting the jacket and cylinde in one piece, but to illustrate jacket design and permit using light metal (as for instance adamite) for the outer casting t reduce weight, a separate jacket casting or "liner" will be used

Before beginning the design ***it is necessary** to fix the maximum pressur to which the parts are subjected*. This is normally 80 lbs., but in order t allow a reasonable margin (to prevent continual blowing of the safety valve) 125 lbs. may be taken as the maximum steam pressure.

Now for 8½ in. diameter of liner, the total load coming on the line (when there is no steam in the jacket), is

$$\frac{8.5}{2} \times 125 = 531 \text{ lbs. per in. of length.}$$

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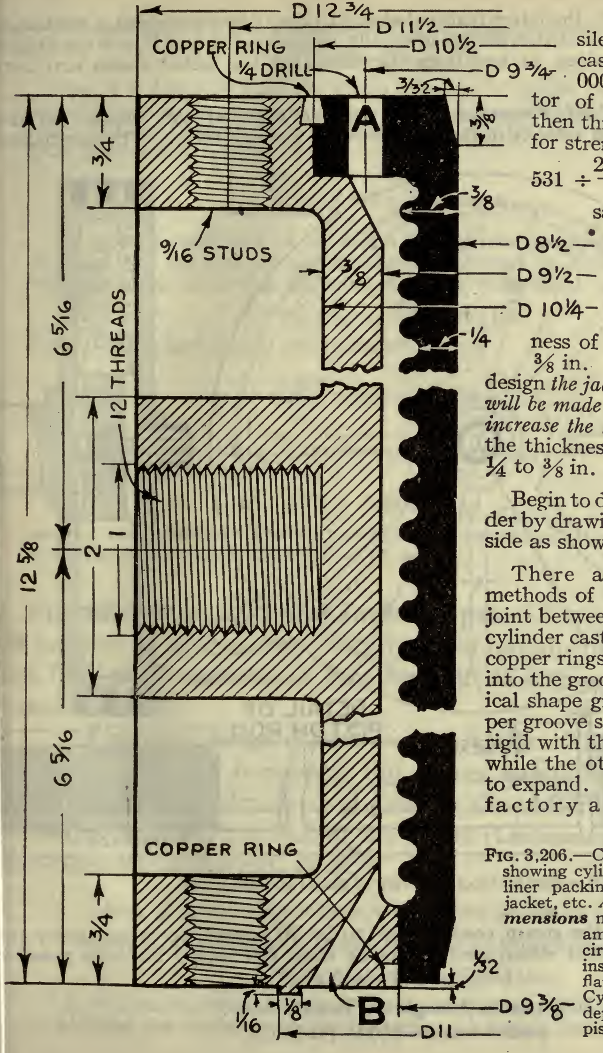
[*] NOTE.—***The large type*** suggests that the designer should fix well in mind the ***cylind dimensions***.



Taking the tensile strength of cast iron at 20,000 lbs. and factor of safety at 10, then thickness of walls for strength is

$$531 \div \frac{20,000}{10} = .27 \text{ in.},$$

say ¼ in.

If provision is to be made for reboring, add ⅛ in. making thickness of liner ¼+⅛=⅜ in. In the present design *the jacket side of liner will be made* ***corrugated*** *to increase the heating surface*, the thickness varying from ¼ to ⅜ in.

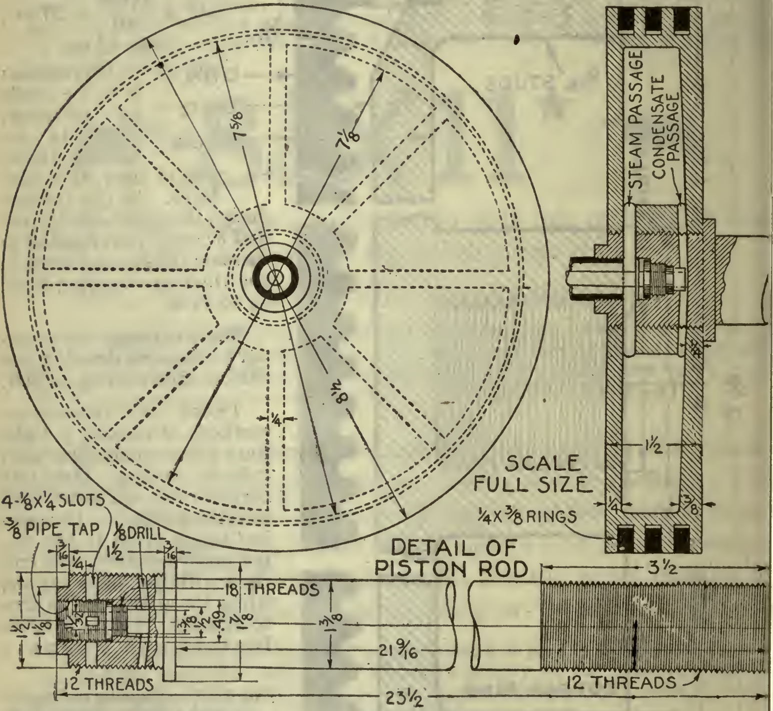
Begin to design the cylinder by drawing detail of one side as shown in fig. 3,206.

There are numerous methods of making a tight joint between the liner and cylinder casting. Here soft copper rings are hammered into the grooves. The conical shape given to the upper groove secures that end rigid with the outer casting while the other end is free to expand. This is a satisfactory and compact

FIG. 3,206.—Cylinder end details showing cylinder casting, liner, liner packing rings, ports to jacket, etc. ***All horizontal dimensions*** marked D, are diameters; thus bolt circle diameter is 11½ ins., diameter liner flange, 10½, etc. Cylinder length which depends on design of piston, is 12⅝ ins.

arrangement, the latter feature helping to keep down the width of the flange which on jacketted cylinders is usually excessive. A and B, are the drilled passages leading to and from the jacket for the jacket steam and condensate.

The Drilled Passages.—The number required will depend on the diameter and velocity of the steam flowing through them. The calculation is as follows:



Figs. 3,207 to 3,209.—Plan and sectional view of piston and piston rod.

Assuming the steam consumption to be 20 pounds per horse power pe hour, then total steam used by engine when developing 30 horse power i

$$20 \times 30 = 600 \text{ lbs.}$$

of which say 10% passes through the jacket, then

$$\text{jacket steam} = 600 \times .1 = 60 \text{ lbs.}$$

The volume of 60 lbs. of steam at 80 lbs., gauge pressure (from table) is

$$4.57 \times 60 = 274 \text{ cu. ft.}$$

Now it is desirable that the steam pressure in jacket be as near boiler pressure as possible, hence to reduce the drop to a minimum, assume a low steam velocity through the drilled passages, say 1,000 feet per minute, then total passage area required is

$$\frac{274}{60 \times 1{,}000} \times 144 = .6576 \text{ sq. in.}$$

To secure a rapid flow of the steam through the jacket so as to promote drainage and prevent air accumulation, it should be made of minimum width, say ⅛ in., giving diameter of jacket walls equal to

$$8\tfrac{1}{2} + 2 \times \tfrac{3}{8} + 2 \times \tfrac{1}{8} = 9\tfrac{3}{8} \text{ ins.}$$

for which the load due to the steam pressure is

$$\frac{9.375}{2} \times 125 = 586 \text{ lbs.}$$

The drilled holes being ¼ in. diam., corresponding to .04909 sq. in. area, then number of holes required is

$$.6576 \div .04909 = 13.4$$

and for very thorough distribution of the steam around the jacket symmetry of layout increase this number to, say 20.

Piston, Piston Rod and Jacket Pipes.—The type of piston to be used is the box form with internal radial ribs which serve not only to strengthen the structure but to augment the jacket heating surface.

The general design is shown in fig. 3,207. The lower internal surface is not horizontal but slopes from the rim to the rod so that the condensate will drain to the center.

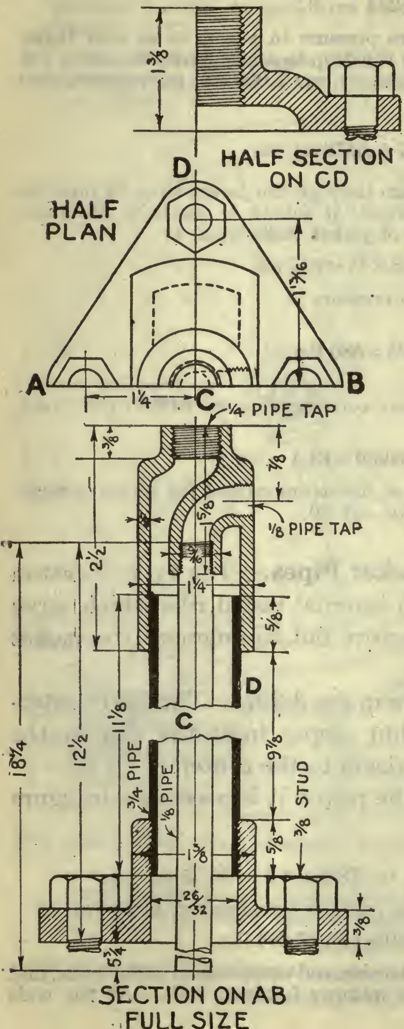
Before designing the hub of the piston it is necessary to figure the diameter of the rod, thus:

$$\text{Area of piston} = \tfrac{1}{4}\pi \times \text{diam.}^2 = .7856 \times 8.5^2 = 56.75 \text{ sq. ins.}$$

$$\text{Load on piston} = \text{area} \times \text{ (steam pressure} + \text{atmosphere pressure)}$$

$$= 56.75 \times (110 + 15) = 7{,}094 \text{ lbs.}$$

The rod has to resist stresses of tension and compression at the ends, and in addition the tendency to buckle midway between ends. At the ends



where only tension and compression is to be considered the stress may for a steel rod be 7,000 lbs. per sq. in., hence for threaded ends minimum area at root of thread = 7,094 ÷ 7,000 = 1.013 sq. ins. Nearest larger diameter corresponding (from table) = 1$\frac{1}{16}$ ins. (area 1.108). Now the diameter of the body of the rod must be greater for stiffness, to resist buckling and this will depend on the ratio of stroke to piston diameter; that is, whether the engine be of the short, medium, or long stroke variety. The stroke being medium, a stress of 5,000 lbs. per sq. in. may be allowed, hence

$$\text{Area of body of rod} = 7{,}094 \div 5{,}000 = 1.42 \text{ sq. ins.}$$

Nearest larger diameter corresponding, from table = 1⅜.

Steam for Piston Jacket.—This is admitted through steam pipe **D** (fig. 3,212) and the condensate removed through the stationary condensate tube **C,** in which slides tube **A,** see fig. 3,208. The size of these tubes are calculated as follows:

$$\begin{aligned} &\text{Area cylinder walls} = \\ &\quad \pi \times 8\tfrac{1}{2} \times 12\tfrac{1}{2} = \quad 333.8 \text{ sq. ins.} \\ &\text{Area cylinder ends} = \\ &\quad 2 \times 56.74 = \quad \underline{113.5} \text{ sq. ins} \\ &\text{Area walls and ends} = 447.3 \text{ sq. ins} \end{aligned}$$

Area both faces of piston = are ends = 113.5 sq. ins., ratio pisto faces : walls + ends = 113.5 : 447.3 or 1:3.9.

FIGS. 3,210 to 3,212.—Details of the ta pipes. Fig. 3,210, half section on **CD** fig. 3,211, half plan on **AB**; fig. 3,21 section on **AB**. The axes referred t correspond to **AB** and **CD** of fig. 3,21

That is, the steam required for the walls and head is 3.9 times that required for the piston, which corresponds to

$$100 \div (1+3.9) = 20.4\% \text{ for the piston, and}$$

$$100\% - 20.4\% = 79.6\% \text{ for the walls and heads.}$$

Now since the total jacket steam is 60 lbs. per hour, or 1 lb. per minute, the total volume of *condensate* to be removed is

$$\frac{1 \times 4.57 \times 12^3}{284.5} \times 1.103 = 27.8 \text{ cu. ins. per minute.*}$$

or, corrected for temperature of steam (323.9° Fahr.)

$$\text{vol. of condensate} = 27.8 \times 1.103 = 30.7 \text{ cu. ins.*}$$

of which 20.4%, or,

$$30.7 \times .204 = 6.26 \text{ cu. ins. per minute.}$$

is to be removed through tube B.

Now for a flow of 25 ft. per minute through tube B,

$$\text{internal area} = \frac{6.26}{25 \times 12} = .02 \text{ sq. ins.}$$

nearest diameter corresponding (from table) = $\frac{5}{32}$ in. (area .0192 sq. in.) and making tube walls $\frac{1}{16}$ in. thick.

$$\text{Outside diam. tube B} = \tfrac{5}{32} + (2 \times \tfrac{1}{16}) = \tfrac{9}{32} \text{ in. } (= .28 \text{ in}).$$

This is just .01 in. *larger* than the inside diam. of a $\frac{1}{8}$ in. pipe, which can easily be machined to fit; bevel, make tube C, $\frac{1}{8}$ in. *pipe* size.

Now volume of steam required for piston jacket is

$$20.4\% \text{ of } 1 \times 4.57 = .93 \text{ cu. ft.}$$

and for a flow of, say 4,000 ft. per minute, the annular area between tubes C and A is

$$\frac{.93}{4{,}000} \times 144 = .034 \text{ sq. in.}$$

$$\begin{aligned} \text{Add to this external area tube C} &= \underline{.129 \text{ sq. in.}} \\ \text{Internal area of tube A} &= .163 \text{ sq. in.} \end{aligned}$$

---

*NOTE.—In the equation, 4.57 is the volume cu. ft. in one lb. of steam at 80 lbs. gauge pressure. 284.5 is the relative volume of water (condensate) at 39° Fahr. The volume of condensate at the temperature of the steam (323.9° Fahr.) is slightly greater in the proportion $62.42 \div 56.58 = 1.103$.

nearest pipe size is ⅜ in. (.191 sq. in. internal area).

Jacket Steam for Heads and Walls.—The volume of steam requir for heads and walls is

$$79.6\% \text{ of } 1 \times 4.57, \text{ or } 4.57 - .93 = 3.64 \text{ cu. ft.}$$

and for a flow of say 4,000 ft. per minute, the annular area between tub D and A is

$$\frac{3.64}{4{,}000} \times 144 = .131 \text{ sq. in.}$$

$$\text{add to this external area tube A} = \underline{.358} \text{ sq. in.}$$

$$\text{internal area of tube D} = .489 \text{ sq. in.}$$

nearest pipe size is ¾ in. (.533 sq. in.).

Jacket Steam Pipe.—The total steam required for all the jackets taken at 10% of that required for the engine or 1 lb. per minute, which 80 lbs. pressure has a volume of 4.57 cu. ft. now for a flow of 6,000 ft. p minute in the jacket steam pipe,

$$\text{internal area} = \frac{4.57}{4{,}000} \times 144 = .1097 \text{ sq. in.}$$

and (from table) diam. nearest pipe corresponding is ¼ in. (internal area .104 sq. in.)

Jacket Drain Pipe.—The total jacket condensate was found to be lb. per minute, of which 79.6% passes through the wall and head jacke to be carried off by the *jacket drain pipe;* that is,

$$79.6\% \text{ of } 1 = .796, \text{ or } .8 \text{ lb.}$$

which (taking the hot condensate at 60 lbs. per cu. ft.) corresponds volume to

$$\frac{1{,}728}{60} \times .8 = 23 \text{ cu. ins.}$$

Now for a flow of say 50 ft. per minute,

$$\text{internal area jacket drain pipe} = \frac{23}{50 \times 12} = .04 \text{ sq. ins.}$$

and from table, nearest (larger) pipe size is ⅛ (having .057 sq. in. area).

Upper Cylinder Head and Valves.—Since the valves ar in the heads, they are to be considered first in designing th heads. The circular space occupied by the piston jacket pipe as designed in figs. 3,207 to 3,209 and the diameters of the valve seats determines the distance of the nearest parts from the center of the cylinder, which in turn fixes the length of the ports.

It should be noted that the valves are **not** *Corliss valves* —they are *rocking valves of the tapered plug type*, the object of using this form being to obtain a valve that will remain tight regardless of wear and one that is practically balanced.

There are an admission and an exhaust valve at each end of the cylinder, located in the heads to secure, 1, practically zero clearance, and 2, a semi-"uniflow" effect, desirable economical features.

The admission valve takes steam from the end and delivers it to the cylinder through a port in its side; the exhaust valve receives steam through the side and exhausts through the end. The valves being opened at each end are practically balanced with respect to the steam, the springs seen at the end press the valves against their tapered seats; this arrangement necessitates stuffing boxes where the steam rods pass out of the valve chambers.

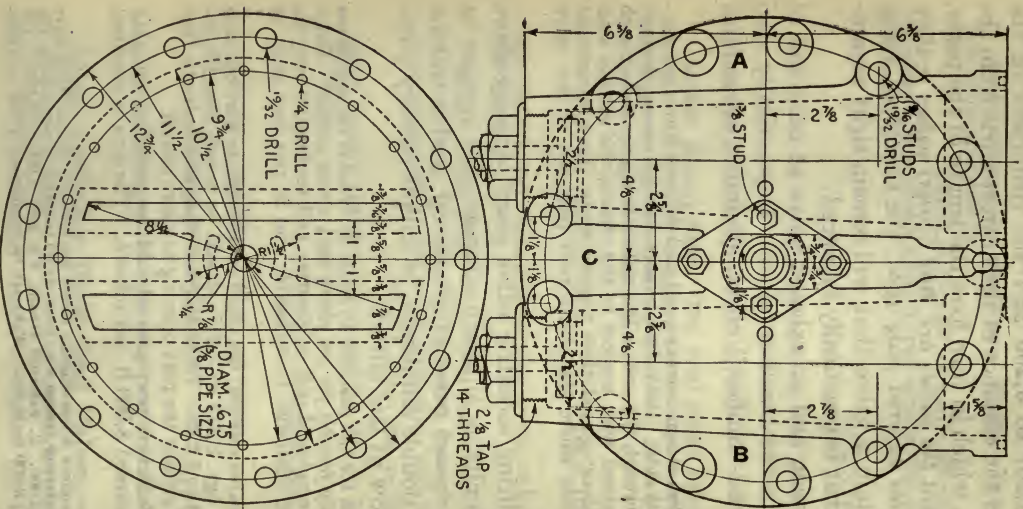
In the design of shifting eccentric valve gears for very short cut offs, the chief difficulty is *to obtain adequate port opening at short cut off without too much travel at late cut off*, and it will be necessary to resort to double admission, or supplementary ports to avoid excessive throw at late cut off.

The first question then for the designer to settle is: *What is the maximum throw that is mechanically feasible for an eleven inch stroke engine?* Since the maximum throw is employed only in starting and backing, it may be made larger than would otherwise be advisable and in this case may be fixed at, say 2¾ ins.

Before designing the valve motion, the effect of inadequate port opening as shown by the cards, figs. 3,212 to 3,217, should be noted, also the influence of lap, figs. 3,218 to 3,220. These Bilgram diagrams indicate that the greater the lap, the greater the port opening at short cut off.

Admission Port.—Sufficient port area should be provided so that the steam flow will not exceed 8,000 ft. per minute. The area is obtained from the formula.

NOTE.—Triple expansion engines where diagram factor lies between .7 and .6 can have the following steam speeds: 1. ***entering steam h.p. ports***, 6,000 to 8,000 ft. per minute; *int.* 7,500 to 10,000; *l.p.* 9,000 to 13,000. 2. ***exhaust steam, h.p. port*** 5,000 to 6,000 ft. per minute; *int*, 6,500 to 8,000; *l.p.*, 8,000 to 10,000. ***Exhaust to condenser,*** 7,000 to 9,000. —*Bragg.*



FIGS. 3,213 and 3,214.—Two plans of upper cylinder head. Fig. 3,214, plan from ***below*** showing, steam and exhaust passages, and the drilled holes; fig. 3,213, plan from ***above*** showing valves, arrangement of studs, and tail pipe connection.

NOTE.—***Cylinder head bolts.*** According to *Whitham*, diameter of bolt circle for cylinder head = diameter of cylinder + 2 × thickness of cylinder + 2 × diameter of bolts. The bolts should not be more than 6 ins. apart. *Marks* gives for number of bolts $b = .7854\ D^2 p \div 5{,}000\ c$, in which c = area of a single bolt; p = boiler pressure lbs. per sq. ins. The safe strain per sq. in., on the nominal area of the bolt is taken at 5,000 lbs.

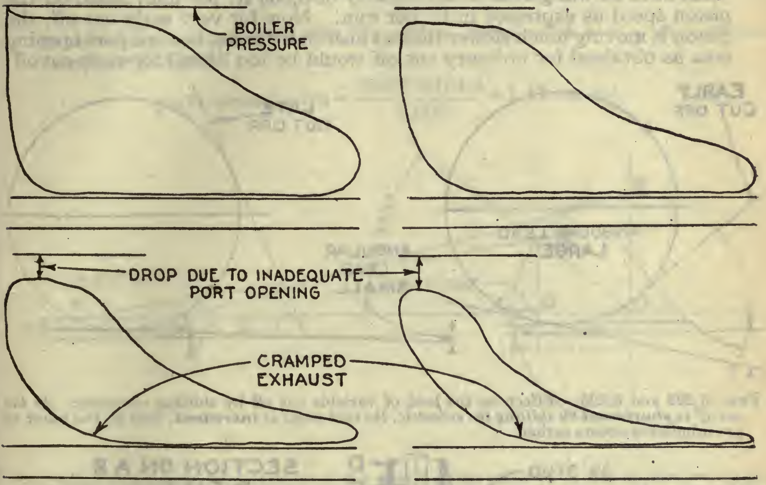
NOTE.—***Thurston says:*** Cylinder flanges are a little thicker than the cylinder and usually of equal thickness with the flanges of the heads. Cylinder bolts should be so closely spaced as not to allow springing of the flanges and leakage, say, 4 to 5 times the thickness of the flanges. Their diameter should be proportioned for a maximum stress of not over 4,000 to 5,000 lbs. per sq. in.

NOTE.—***Seaton*** objects to webs for stiffening cast iron cylinder covers as a source of danger. The strain on the web is one of tension and if there should be a nick or defect in the outer edge of the web, the sudden application of strain is apt to start a crack.

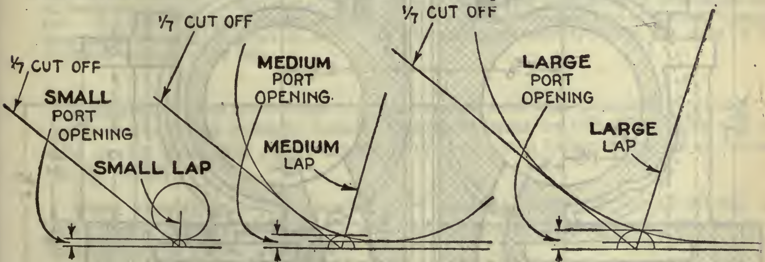
$$\text{area} = \frac{\text{piston area} \times \text{piston speed}}{\text{steam flow}} \quad \dots\dots\dots\dots\dots (1)$$

Piston area $= .7854 \times (8\frac{1}{2})^2 = 56.75$ sq. ins.; piston speed $= 2 \times 250 \times {}^{11}/_{12} = 458$ ft.; steam flow $= 8{,}000$ ft. Substituting these values in (1)

$$\text{Area} = \frac{56.75 \times 458}{8{,}000} = 3.25 \text{ sq. ins.}$$



FIGS. 3,215 to 3,218.—Indicator cards of "automatic cut off" engine showing effect of the shifting eccentric on admission at early cut off, ***resulting in drop due to inadequate port opening***. The cards also show at early cut off ***cramped exhaust due to premature closing of the exhaust valve***, illustrating the need of separate independent exhaust valves.

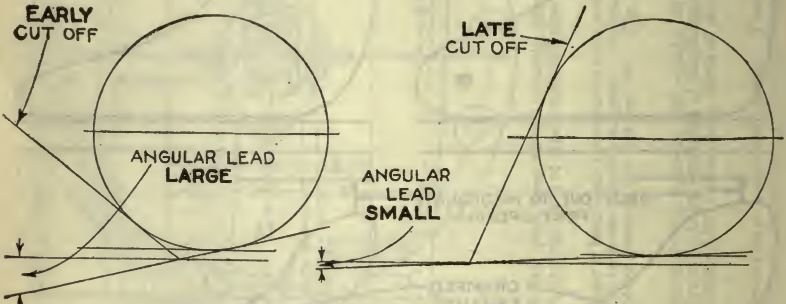


FIGS. 3,219 to 3,221.—Bilgram diagrams illustrating influence of lap on the port opening for early cut off. It is evident from the diagrams that ***the greater the lap the greater the port opening at early cut off***.

Exhaust Ports.—The velocity of steam flowing out of the exhaust port will be limited to 4,000 ft. per minute to reduce back pressure to a minimum. The port area then will be twice the admission port area, or,

$$3.25 \times 2 = 6.5 \text{ sq. ins.}$$

Port Opening Area.—For ordinary cut offs the area of port opening is made such as will give a steam velocity of 8,000 ft. per min. based on the piston speed as expressed in ft. per min. Now for very early cut off, the piston is moving much slower than at near half stroke, hence a port opening area as obtained for ordinary cut off would be too liberal for early cut off,



FIGS. 3,222 and 3,223.—Effect on the lead of variable cut off by shifting eccentric: *As the cut off is shortened by shifting the eccentric, the lead angle is* ***increased***, that is, the point of pre-admission occurs earlier.

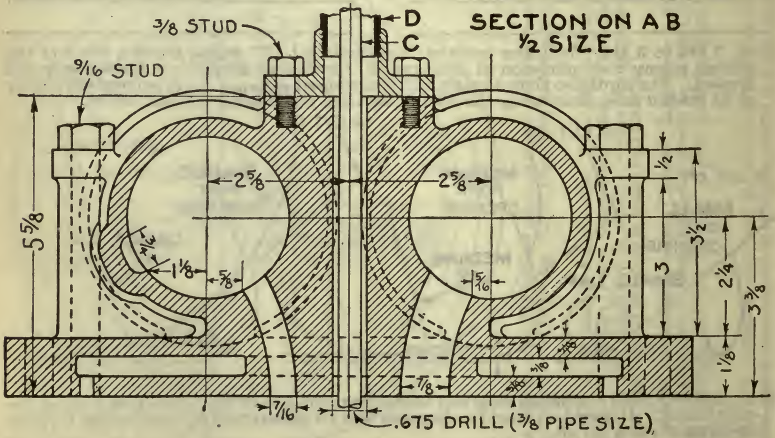


FIG. 3,224.—Elevation of upper cylinder head (section on **AB** of fig. 3,214).

and in fact would be beyond the limit of the valve gear. Hence the aim will be to secure a flow of 8,000 ft. per min. referred to the actual velocity of the piston at the point of cut off. Accordingly at the point of $^1/_7$ cut off, the piston is moving at a velocity of 164.4 ft. per min., as obtained in fig. 3,225. To figure the area required,

$$\text{port opening area} = \frac{\text{area piston} \times \text{piston speed}}{\text{steam velocity}} \qquad (1)$$

Substituting, piston area = 56.75; piston speed = 164.4 ft., and steam velocity = 8,000 ft.,

$$\text{port opening area} = \frac{56.75 \times 164.4}{8{,}000} = 1.17 \text{ sq. ins.}$$

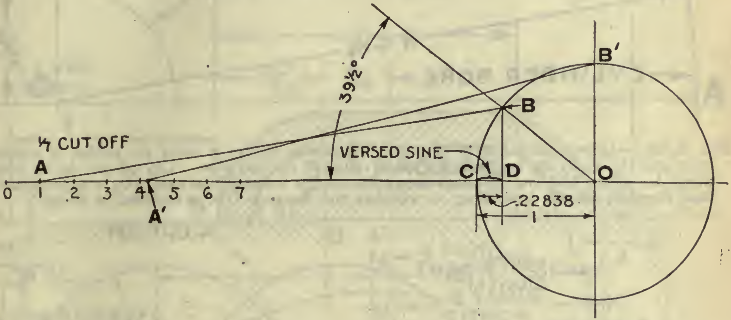


FIG. 3,225.—Instantaneous velocity of the piston at the point of $^1/_7$ cut off (connecting rod ratio 2:1). Lay off connecting rod position A B for $^1/_7$ cut off giving crank angle COB = 39½° as measured by protractor, and position A′ B′ for crank angle of 90°. OB′ is crank position of maximum piston velocity because at this instant the crank pin and wrist pin are moving in parallel paths, and the actual velocity of the piston at this instant = velocity of crank pin = (34.558 × 250) ÷ 12 = 720 ft. per min. neglecting angularity of the connecting rod. Now assuming the velocity of the piston to vary as the versed sine of the crank angle, then piston velocity at point A, or $^1/_7$ cut off = maximum velocity × ver. sin. 39½° = 720 × .22838 = 164.4 ft. per min.

Length of Ports.—Since the piston rod and tail circulation pipe projects through the cylinder heads, the ports must be offset on each side of the center line, the approximate length of ports being 8 ins. as obtained in fig. 3,226.

Port Opening.—The area of port opening required for admission as just found being 1.17 sq. ins., and the approximate length of port being 8 ins. then the port opening is

1. For single port = 1.17 ÷ 8 = .146 in.
2. For double ports = 1.17 ÷ 16 = .073 in.

Width of Ports.—*Divide area of port by the length.* From values just found,

width of single admission port = 3.25 ÷ 8 = .406, say $\frac{7}{16}$ in.
width of exhaust port = 6.5 ÷ 8 = .813, say $\frac{7}{8}$ in.

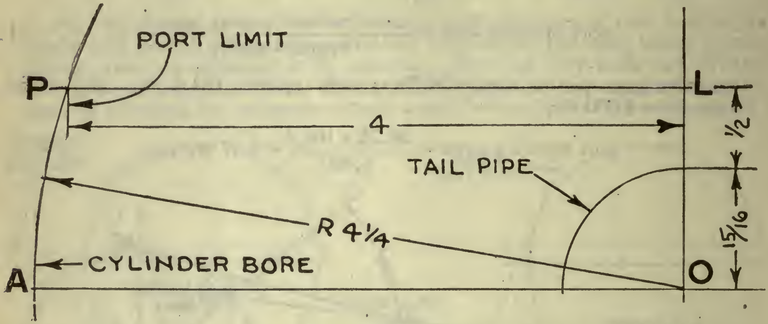


FIG. 3,226.—Approximate length of ports. Describe an arc R = 4¼ in. radius of cylinder bore and the small arc representing space required for tail pipe. Lay off say ½ in. giving point L. Then the line PL, drawn parallel to the longitudinal axis AO, is approximately half the port length depending on whether the distance OL, is retained in design.

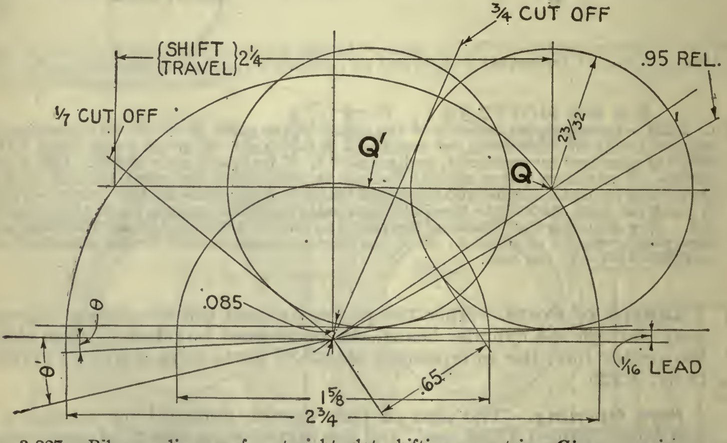


FIG. 3,227.—Bilgram diagram for straight slot shifting eccentric. ***Given:*** maximum cut off ¾; minimum cut off $^1/_7$; lead; $\frac{1}{16}$. ***By measurement:*** Lap $\frac{23}{32}$; port opening at ¾ cut off, .65; port opening at $^1/_7$ cut off, .085. It should be noted that in the operation of the straight slot eccentric under the conditions of the above diagram, the angle of pre-admission changes from θ to θ' as the center of the eccentric moves from Q to Q′ in shortening the cut off from ¾ to $^1/_7$.

Admission Valve and Seat.—In fig. 3,227, the Bilgram diagram* is constructed in the usual way having given the following data: connecting rod ratio 2:1; cut off, max., $\frac{3}{4}$ min., $^1/_7$; lead $\frac{1}{16}$; maximum throw $2\frac{3}{4}$ ins. The solution gives: lap, $\frac{23}{32}$; port opening, at $\frac{3}{4}$ cut off, .65 (max.) and at $^1/_7$ cut off, .085 (min.) as marked on the diagram.

To get dimensions of valve and seat just lay out as for an ordinary slide valve as in fig. 3,228. Since the rocking form of valve is used, the next problem is to determine the diameter of the valve seat. Now in order to secure quick movement of the valve during admission, a wrist plate motion will be used, the effect of which is to make the travel of the valve less than

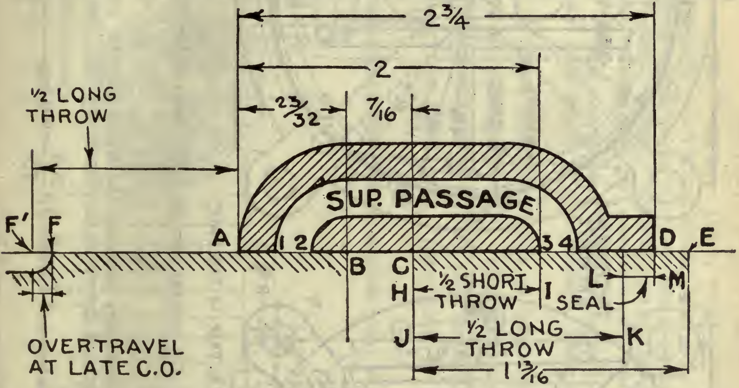


FIG. 3,228.—Preliminary layout of valve and seat. Draw steam port BC = $\frac{7}{16}$ in. and lay off steam lap AB = $\frac{23}{32}$ in.; this lap contains the supplementary port 1,2, which is located at the middle making A1, and B2, each equal say $\frac{1}{4}$ in. The supplementary port is made larger than necessary to offset friction. To prevent port BC, being uncovered (at short cut off) except during admisson lay off HI equal $\frac{1}{2}$ short throw and make C3 at least this distance. Draw supplementary passage terminating at 3,4. The seat limit E is found by making 4E = steam lap AB. To find end D of valve, lay off JK = $\frac{1}{2}$ long throw and LM the amount of seal desired say $\frac{1}{4}''$. To locate the seat limit at F, locate F′, at a distance from A = $\frac{1}{2}$ long throw, and make F′ F equal to the desired overtravel. F should be so located that the supplementary passage does not overtravel it, otherwise there would be at second admission after cut off. If in design the seat should extend all around the valve the depression at F, is not necessary.

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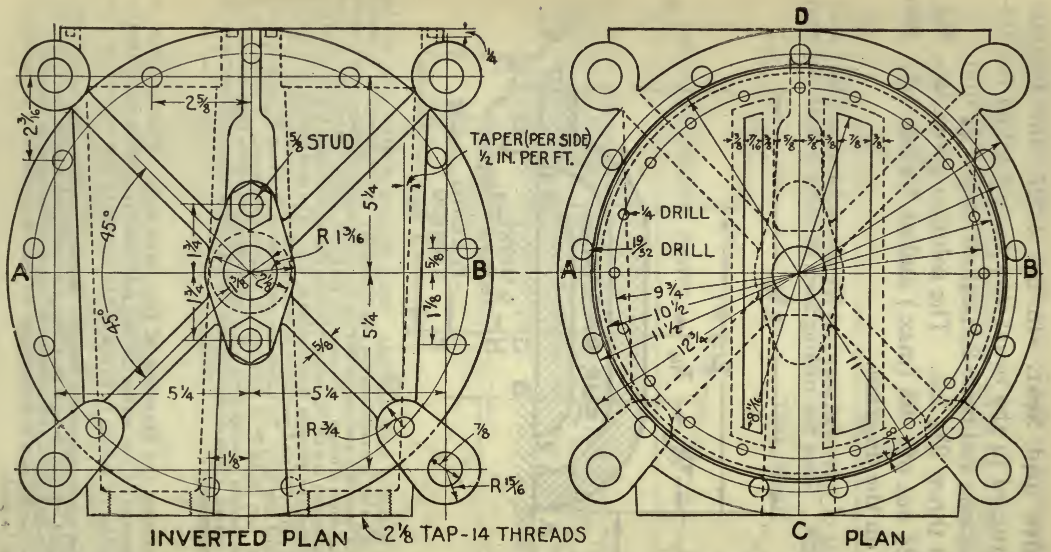
*NOTE.—A thorough knowledge of the Bilgram diagram, which is explained in great detail in Chap. 4, is assumed. It will be seen that for $^1/_7$ cut off and double ports, the port opening .085 in. is ample, being .085 − .073 = .012 in, in excess of the amount required. The equivalent of double ports is obtained with simplicity by the use of the Allen or supplementary port valve which will be used.



INVERTED PLAN

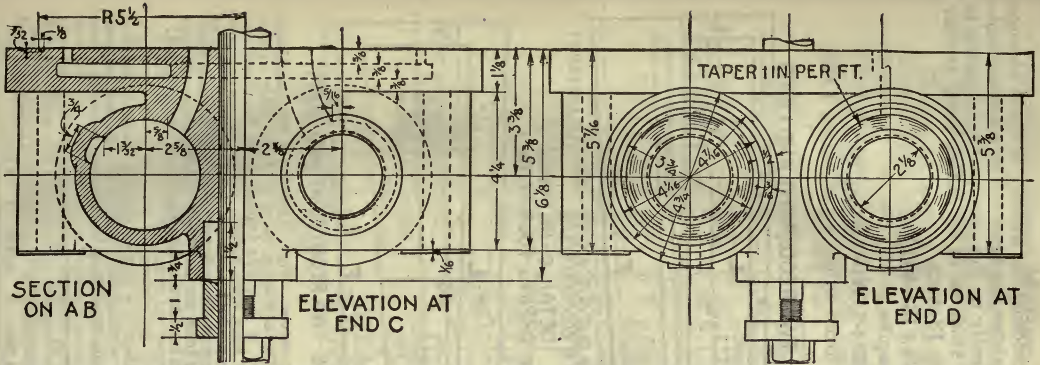
PLAN

Figs. 3,229 and 3,230.—Lower cylinder head, The inverted plan fig. 3,229, shows plainly the four projecting feet to which are attached the four upright columns of the frame.

the throw of the eccentric, and for preliminary purposes the travel may be assumed to be reduced from $2\frac{3}{4}$ ins. to say 2 ins.; and the steam arm of the valve to vibrate say 75°, then

$$\text{circumference of valve} = 2 \times \frac{360}{75} = 9.6 \text{ ins.}$$

diameter corresponding (from table) $3\frac{1}{16}$ ins., say 3 ins.



FIGS. 3,231 and 3,232.—Lower cylinder head. Fig. 3,231, elevation at end C.; fig. 3,232, elevation at end D.

Lower Cylinder Head.—This is practically the same as the upper head with the exceptions of projecting feet to receive the frame columns and a stuffing box for the piston rod instead of the tail pipe opening. The stuffing box is designed to accommodate ⅜ in. packing and is amply deep to secure a tight joint.

The groove turned in the face of the head registers with a similar ring projecting from the cylinder flange and is to facilitate centering the cylinder with the head in erecting.

Cylinder Head Studs.—In order to avoid too large and thick flanges the studs should not be any larger than necessary. The load to be carried on the studs is usually based on the area of the stud circle, but with a jacketted cylinder, because of the abnormal width of the flange, a less liberal allowance will suffice, figuring in this case to the liner flange circle which (from fig. 3,206) is 10½ in. in diameter,

$$\text{area } 10\tfrac{1}{2} \text{ circle (from table)} = 86.59 \text{ sq. ins.}$$

Load for 125 lbs. steam pressure = 86.59 × 125 = 10,824 lbs.

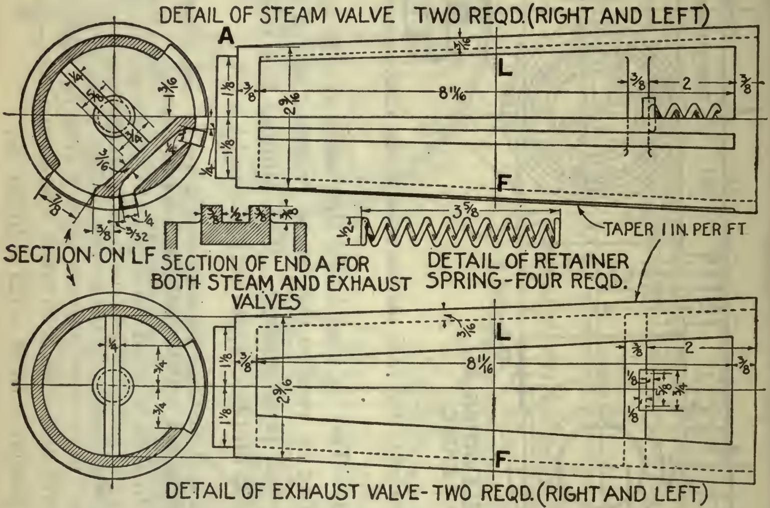
A difficulty is here encountered in choosing the number of studs because of the valve chambers which project above the head. The studs are normally spaced equally apart, but evidently in this case uniform spacing is impossible, the arrangement with 13 studs as in fig. 3,214, being as near uniform as can be obtained. Now with 13 studs,

$$\text{load per stud} = 10{,}824 \div 13 = 833 \text{ lbs.}$$

For a load on stud of 5,000 lbs. per sq. in.,

$$\text{area at root of thread} = 833 \div 5{,}000 = 1.67 \text{ sq. in.}$$

diameter nearest stud corresponding (from table) $\frac{9}{16}$ in. (area **1.62**).


DETAIL OF STEAM VALVE TWO REQD.(RIGHT AND LEFT)
A
L
F
8 11/16
2 9/16
1 1/8
3/8
2
3/16
1/4
7/8
3/32
5/32
SECTION ON LF
SECTION OF END A FOR BOTH STEAM AND EXHAUST VALVES
1/2
3 5/8
DETAIL OF RETAINER SPRING-FOUR REQD.
TAPER 1 IN. PER FT
3/4
5/8
1/8
DETAIL OF EXHAUST VALVE-TWO REQD.(RIGHT AND LEFT)

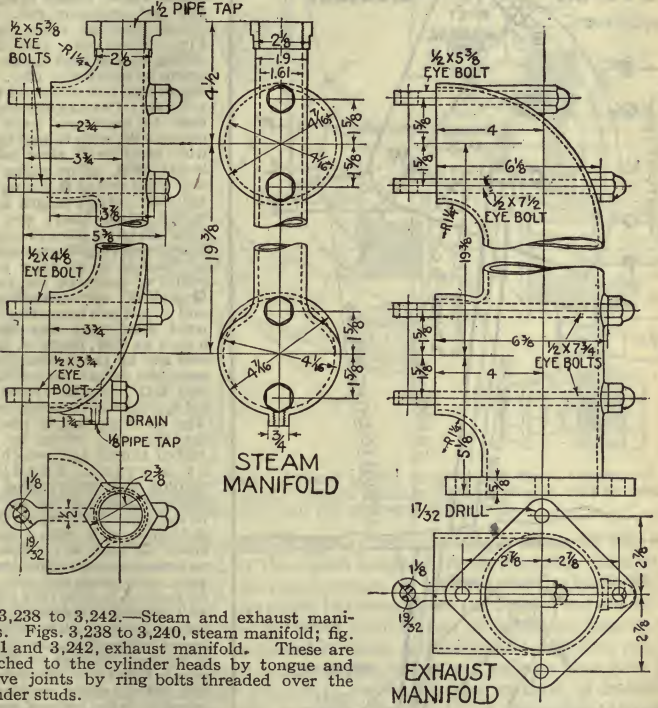
Figs. 3,233 to 3,237.—Admission and Exhaust valves. Figs. 3,233 to 3,235, admission valve; figs. 3,235 and 3,237, exhaust valve. The admission valve is of the Allen or supplementary port type, giving double admission, a desirable feature with the short cut off (see pages 208 to 210 of vol. I for design of Allen slide valve). Both valves are open at both ends to render them balanced. The two projections at the end engage with the lug on steam and exhaust spindle, so that the angular movement of the spindle will be transmitted to the valve. The admission valve takes steam from the large end and discharges through the side. The exhaust valve takes steam through its side and exhausts from the large end.

NOTE.—***Stress on cylinder bolts.***—Since there is a proportionately greater straining action on small bolts due to screwing up, a smaller working stress should be adopted for small cylinders than for large cylinders. Thus the working stress may be taken according to *Unwin* as follows:

Diameter of cylinder up to.......	10	15	20	60	60 ins. and greater
Stress, in lbs. per sq. in.........	3,000	4,000	5,000	6,000	7,000

The Wrist Plate Motion.—The effect of the wrist plate is *to* ***distort*** *the valve movement so that it will move* ***fast*** *while opening and closing, and* ***slow*** *after closing.*

This rapid opening and closing reduces wire drawing which gives sharp admission and cut off corners on the indicator diagram thus increasing the efficiency. Moreover, the reduced or slow movement of the valve during the closed period when the valve is most unbalanced reduces the amount of work necessary to operate the valve during this period.



FIGS. 3,238 to 3,242.—Steam and exhaust manifolds. Figs. 3,238 to 3,240, steam manifold; fig. 3,241 and 3,242, exhaust manifold. These are attached to the cylinder heads by tongue and groove joints by ring bolts threaded over the cylinder studs.

NOTE.—The cylinder studs and bolts should be made of the best steel, and of such size that the strain on them does not exceed 5,000 lbs. per sq. in. of section at the bottom of the thread, as they are subject to severe and sudden shocks when priming occurs, and to considerable wear and tear from the frequent removals of the covers for examination of the piston. In large cylinders 6,000 lbs. strain is permissible. If iron be used, the strain should be 20% less.—*Seaton.*

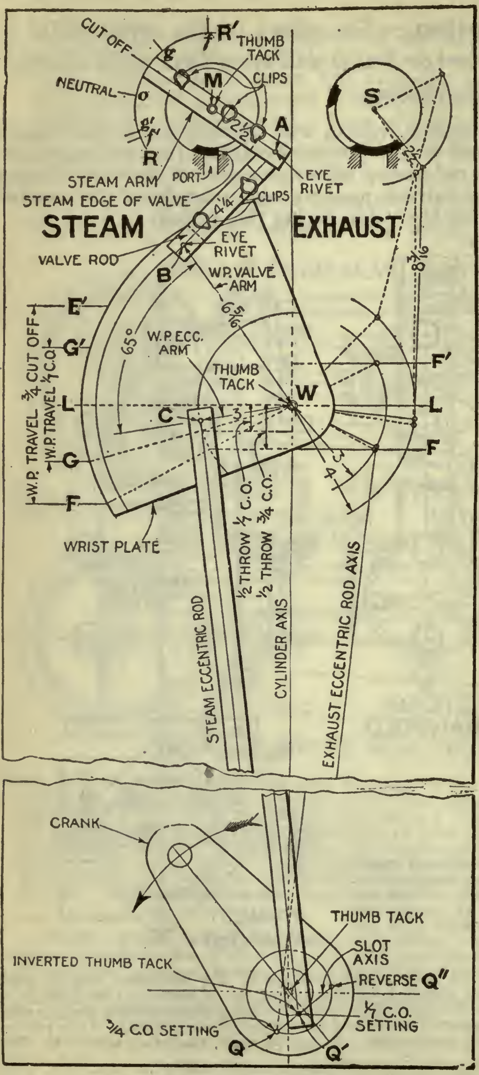
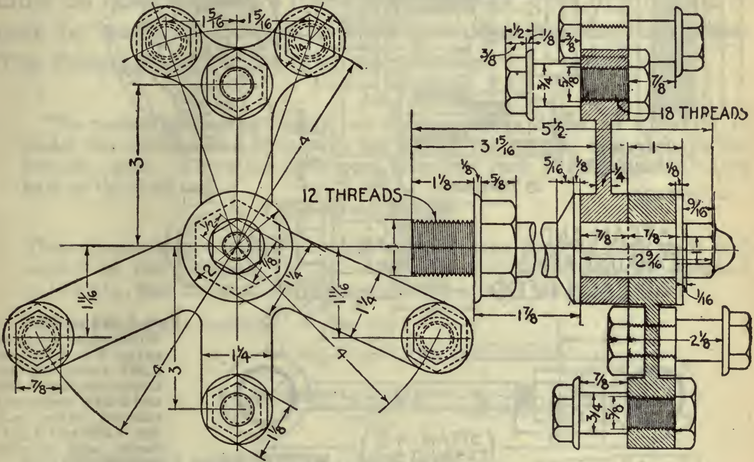


FIG. 3,243.—Cardboard model for design of admission valve gear and diagram of exhaust gear. ***In construction,*** get a large sheet of three ply bristol board, draw cylinder axis and horizontal line **LL**, the intersection **W** giving the center of the wrist plate pin. By aid of fig. 3,224, locate **M** and **S**, centers of admission and exhaust valves respectively. Draw circular valve seat and outline of parts as shown. Cut cardboard discs and strips to represent valve, steam arm, valve rod and eccentric rod. Ink in steam edge of valve on disc. Pivot steam arm and disc by thumb tack **M** and fasten disc to steam arm with a clip. Pivot steam arm and valve rod with an eye rivet. Cut a cardboard quadrant to represent wrist plate; pivot at **W**, with thumb tack and with ***trial wrist plate valve arm*** length **WB**, pivot valve rod to *w.p. valve arm* at **B**, (using thumb tack till correct length is determined, then insert eye rivet). Now cut out a piece of cardboard to represent crank arm and pivot by thumb tack, after locating center of eccentric for positions **Q** and **Q'**, corresponding to ¾, and $^1/_7$ cut offs respectively. On wrist plate take any desirable length as **CW**, for *w.p. eccentric arm* and by the dotted line construction locate horizontals **FF'** and **GG'**, angular travel limits for ¾ and $^1/_7$ cut offs respectively. The eccentric rod is pivoted to the eccentric center by an inverted thumb tack which serves as a crank handle for turning the gear, being easily changed from one cut off setting to the other by changing position of thumb tack with respect to holes **Q** and **Q'**. Now *make various alterations of the lengths of parts, with crank and valve in cut off position*, noting that adequate *port opening* is obtained, and for these various alterations note travel of steam arm on either side of neutral position as indicated on arc **RR'** by *of*, *of'*, (for ¾ cut off) and *og* and *og'*, for $^1/_7$ cut off, calculating the ratio of each pair of values thus found. The author after numerous trials obtained maximum ratio of 2.38 for ¾ *c.o.* and 1.48 for $^1/_7$ *c.o.* with the dimensions as marked on the model. The diagram of the exhaust gear is constructed by similar methods.

The wrist plate mechanism comprises the steam arm, wrist plate arm, and steam valve rod and the problem here encountered consists in finding lengths of these parts and angular relations that will give the maximum wrist plate effect. It may be solved by taking trial lengths of the parts and drawing a diagram showing neutral and extreme positions of the motion, or by making a card board model and altering the various parts until the most desirable motion is obtained. In either case the neutral or mid-position of the model should be drawn (or model put in this position) and the ratio of the valve arm throws examined. Average values for this ratio, which is a measure of the "wrist plate effect" are 2.5 to 3.



FIGS. 3,244 and 3,245.—Wrist plates. Instead of the usual heavy discs. consists of arms, thus avoiding any unnecessary metal. Before making this drawing it was necessary to locate the position of the eccentrics to find the overhang of the arms.

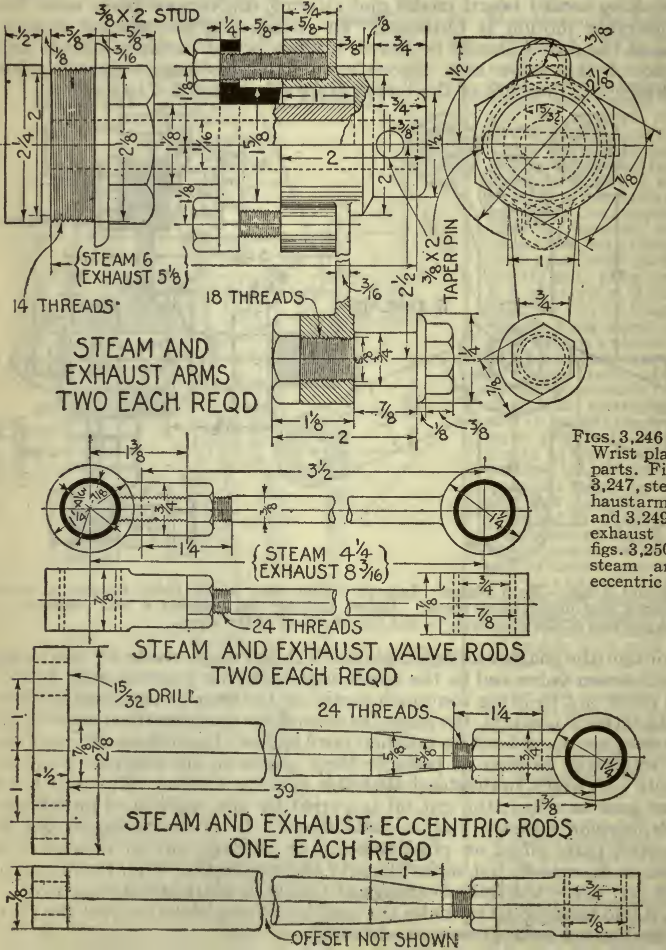
To obtain the maximum wrist plate effect the angle made by the steam arm and steam valve rod in the small angle extreme position *must be 90°*, and in order not to bring too much strain on the gear in the large angle or *stretched* extreme position, the angle between these parts *must not exceed 140°*.

The designer should not be satisfied until he has obtained an arrangement which will make as near as possible these angles in the extreme position. It should be clearly understood that this is not a Corliss valve gear but a positive gear in which the cut off is varied by the method of ***combined variable angular advance and variable throw.*** It is desirable to get as much wrist plate effect as possible at 1/7 or working cut off instead of at full gear or ¾ cut off, but unfortunately the greatest wrist plate effect is at late cut off, hence the full advantage of the wrist plate is not obtained.

Fig. 3,243 shows how to make the card board model and adjust the parts until the desired arrangement is obtained.

The steam arms, exhaust arms and their connections with the wrist plates are similar to a double eccentric Corliss gear, but it should be understood that this is a ***non-releasing gear*** (hence there are no dash pots) the cut off being varied by the method of ***combined variable angular advance, and variable throw.***

The chief feature of the valve gear is the way in which this is worked out mechanically and the valves themselves.



Figs. 3,246 to 3,253.—Wrist plate motion parts. Fig. 3,246 and 3,247, steam and exhaust arms; figs. 3,248 and 3,249, steam and exhaust valve rod; figs. 3,250 and 3,251, steam and exhaust eccentric rod.

Bed Plate.—Except in extreme light weight engines the bed plate should be made very substantial, not only for strength, but to help absorb the vibrations of the moving parts and thus prevent them being transmitted to the boat. Accordingly the bed plate is made ⅛ inch thicker than the cylinder castings. Before the drawing can be completed the air and feed pumps must be designed and a "layout" of center lines of their drive gear be made to determine their positions on the bed plate. The design should be as open as possible.

The author's favorite design, as shown in figs 3,254 to 3,256 is to make the shaft center flush with the top of the casting and to have the bottom open. There are two cross members each of which must carry half of the load, or

$$7{,}094 \div 2 = 3{,}547 \text{ lbs.}$$

These cross members may be regarded as beams fixed at both ends and loaded in the middle. In fig. 3,256 the middle section as designed is ½×4¼, and the span (between columns) is 10½ ins.

Now, using the formula $P = \frac{4}{3}\,\frac{R\,bd^2}{l}$ and substituting

$$P = \frac{4}{3} \times \frac{50{,}000 \times \frac{1}{2} \times (4\frac{1}{4})^2}{10\frac{1}{2}} = 57{,}341 \text{ lbs.}$$

and the factor of safety $= 57{,}341 \div 3{,}547 = 16$
which is increased by the top flange and is ample.

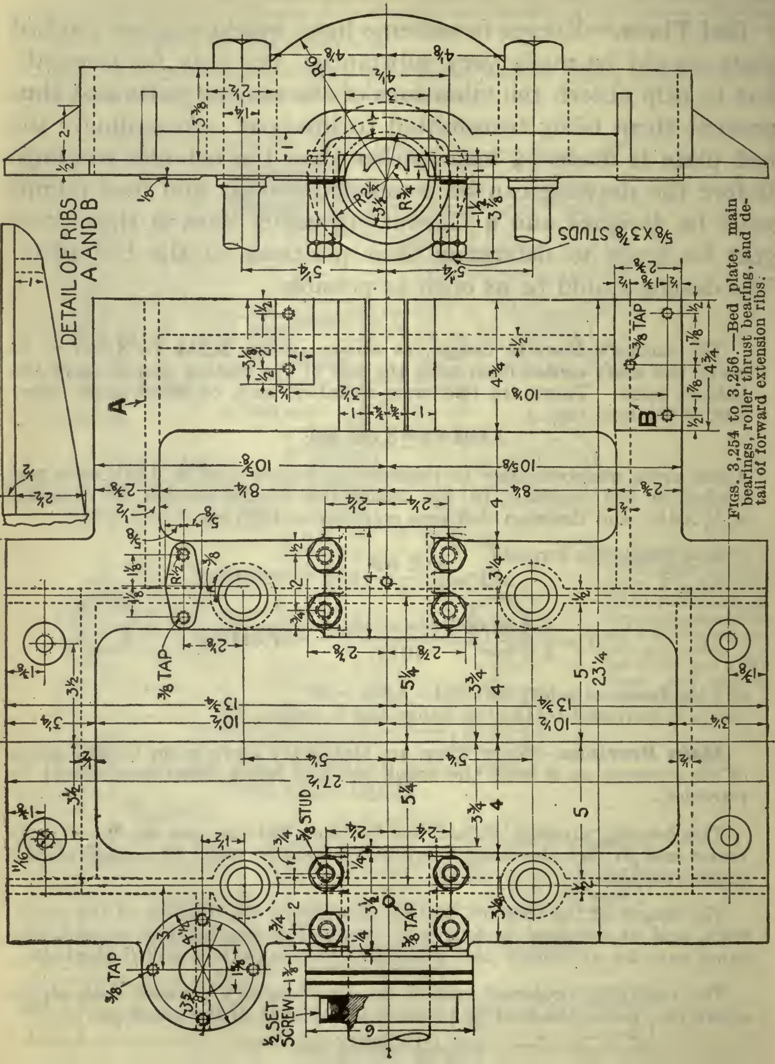
Main Bearings.—Since these are stationary there is no cooling effect of air currents as is with the crank pin and hence more area should be provided.

The bearing pressure should not be over 300 lbs. per sq. in.—less is better and in fact the main bearings are usually made to occupy all the space available.

The length of the forward bearing depends on the location of the eccentrics, and as designed is four ins. long. The aft bearing is made the same size, for symmetry and in order that one pattern will do for both.

The combined projected area of the two bearings is twice that of the crank pin, hence the bearing pressure is half that of the crank pin or

$$591 \div 2 = 295\frac{1}{2} \text{ lbs.}$$



FIGS. 3,254 to 3,256.—Bed plate, main bearings, roller thrust bearing, and detail of forward extension ribs.

which is within the 300 lbs. limit. Four studs are used for each bearing, their size being obtained as follows:

load carried by each stud $= \frac{1}{8}$ of total load on piston
$= \frac{1}{8}$ of $7{,}094 = 887$ lbs.

For a stress of 5,000 lbs. per sq. in. on stud.

Area at root of thread $= 887 \div 5{,}000 = .178$ sq. in.

nearest larger size stud (from table) is $\frac{5}{8}$ in. (area at root .202 sq. in.).

Each stud is provided with two nuts to securely fasten the cap. The bearings are lined with either Babbitt metal or some other (so-called) anti-friction metal. Experience teaches that a bearing so lined will run cooler, probably because the lining metal being softer, accommodates itself to the journal more readily than a harder metal.

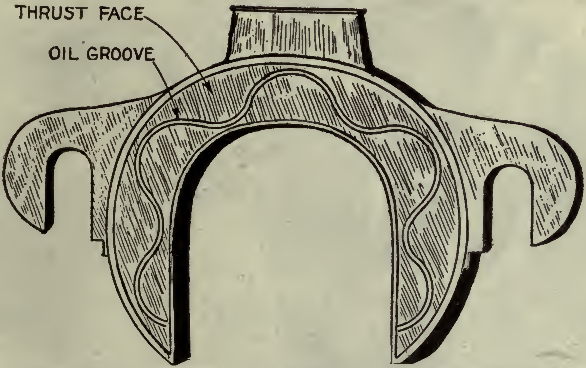


Fig. 3,257.—Old style cast iron horse shoe thrust bearing—*could anything be worse?* It requires a water jacket and unlimited supply of oil to keep it cool.

As To Thrust Bearings.—Is not the cost of power high enough at the present price of coal *without putting a brake on the engine?* This is virtually what is being done by the use of the old fashioned thrust bearing. Why are they used? Someone within the memory of the oldest inhabitant devised one of these mechanical outrages and everyone followed the suggestion. Any designer, worthy of the title, should think seriously before adding another one of these relics of the dark ages to the large number now in use which should never have been made.

In the present state of perfection of the art of making roller bearings of all kinds, what can be the objection to the use of a roller thrust bearing?

Most engineers will say they are all right for small work, but are afraid to use them on large engines.

Why?

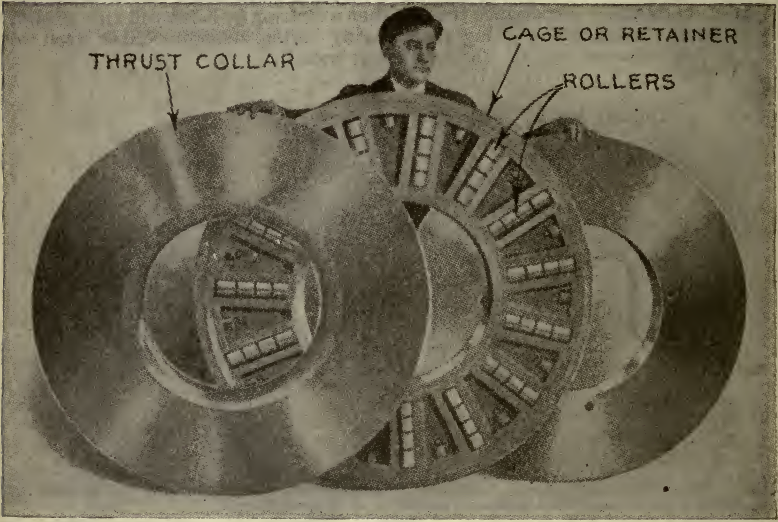


FIG. 3,258.—Plain roller thrust bearing. Weight complete 7,500 lbs.; load carried 1,500,000 lbs., running at 100 revolutions per minute,

Fig. 3,258 ought to inspire confidence. As to saving in coal, oil and attendance, the author does not believe the following letters, taken from the catalogue of a prominent manufacturer, in any way overrate the results obtained:

"In reply to your letter as to the result of the roller bearing thrust which you installed on the yacht "Anita" last April, will say it has proven very satisfactory in every respect. We ran about six months the past season, and I did not have the least trouble in any way, and this fall when I had it apart I could not see as it has worn any. We ran 15 hours on ¼ quart of oil, and then the chill was hardly off it. Can also make about four revolutions per minute more than with the old thrust."

"The roller thrust bearing worked excellently and we got six revolutions more out of the engine with the same cut off, than before it was put in. not saying anything about the saving in coal. We have run about 11,000 miles, and have put no oil in thrust bearings since we started, and as for care, the oilers have almost forgotten that there is a thrust bearing in the ship. The saving in the coal on the "Aphrodite" was between four and five tons per day."

Don't design a roller thrust bearing, but design the shaft and bed plate end to accommodate a standard bearing made by some reliable manufacturer. He has had the experience and knows what is required.

Frame.—In small engines especially accessibility to the moving parts is a very desirable feature, and this is secured to a maximum degree by use of the open column type of frame. The frame proper consists of four steel columns, to which is attached the guides, the two on the port side being bored to receive the bearing for the air and feed pump walking beam shaft.

The diameter of the columns is proportioned to secure rigidity as well as strength and the author considers the diagonal braces sometimes used totally unnecessary, especially in small engines as a very rigid structure can be made without unduly increasing the diameter of the columns.

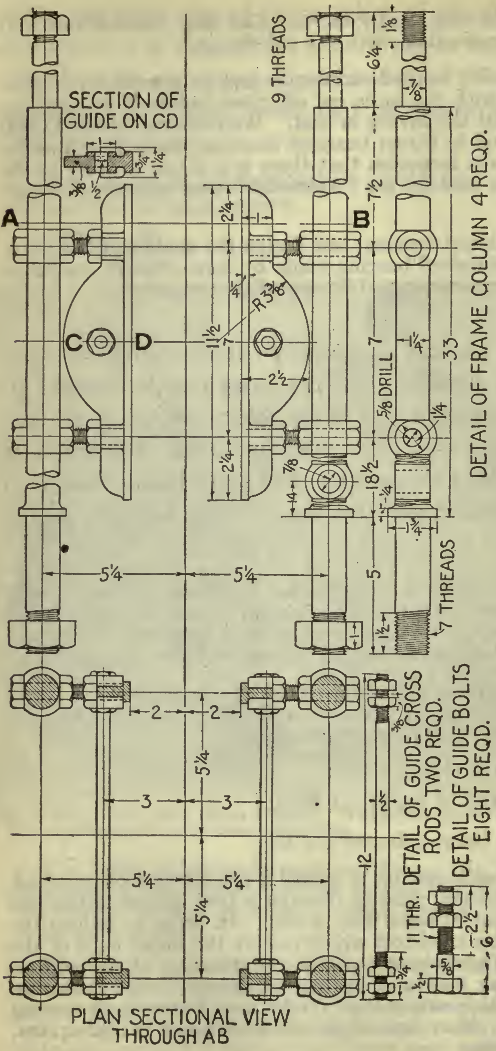
For ordinary piston diameter and stroke ratios, if the diameter of frame columns be made 90% of piston rod diameter it will give an amply rigid frame without any transverse or diagonal bracing.

Now diameter of piston is $1\frac{3}{8}$ in., and

$$90\% \text{ of } 1\tfrac{3}{8} \text{ in.} = 1.24 \text{ in.}$$

hence, make diameter of frame columns $1\frac{1}{4}$ ins.

Near the lower end of each column is turned a projecting footing which rests on the bed plate, the end passing through a hole drilled in the bed plate and secured by a nut of same size as rod. In order to reduce the length of the lower cylinder head feet which receive the upper ends of the columns and thus bring them closer together the diameter of the sections of column is reduced being just large enough for strength, this reduction giving a shoulder upon which wrests the cylinder head feet without turning a projection at this point. Now, area of piston rod as found is 1.42 sq. ins. and there being four columns



¼ of 1.42 = .355 sq. in. which take for area at root of thread at reduced section of column.

Nearest (larger) diameter corresponding (from table) is ⅞ in.

In designing the columns, they should be enlarged at the points where they are bored for the guide bolts and pump walking beam shaft bearings.

Figs. 3,259 to 3,264 show design of frame and guides. Owing to the unusual height of the engine the author has provided diagonal braces for the frame as shown on page 1,726.

Guides.—Four guides are used, two for forward motion and two for reverse. Since a roller bearing is used, the

Figs. 3,259 to 3,264.—Engine frame, guides, and details of guide cross rods and guide bolts. This is the most accessible type of frame, a desirable feature especially in small engines. Note method of adjusting guides to the rollers.

length of guides is made equal to the length of stroke plus a small margin (¼ in.) at each end or total of 11½ ins.

The guides are carried by the frame columns, being attached by two bolts each. These bolts are located, not at the ends, but are spaced nearer the center to reduce the span and thus the size of the guide.

For strength they are designed thus:

At 125 lbs. steam pressure, load on wrist pin (=load on piston) = 7,094 lbs. The maximum thrust on the guides is equal to the total load on wrist pin × tangent of maximum angle connecting rod makes with cylinder axis. The value of the tangent depends upon the ratio of connecting rod to stroke, as follows:

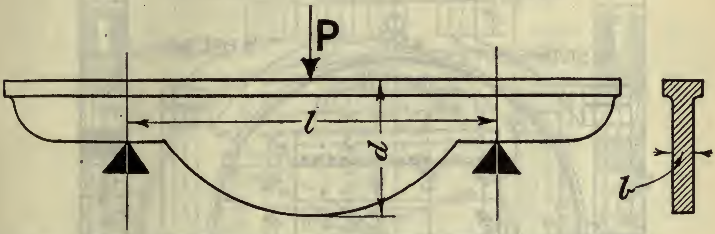


FIG. 3,265.—One of the guides considered as a beam in designing for strength.

connecting rod stroke ratio	2	2½	3
tangent of maximum angle	.258	.204	.169

The length of connecting rod is 22 ins. and stroke 11, hence connecting rod stroke ratio is

$$22 \div 11 = 2$$

Substituting in the equation just given

$$\text{thrust on guides} = 7{,}094 \times .258 = 1{,}830 \text{ lbs.}$$

and since this is carried by two guides

$$\text{thrust per guide} = 1{,}830 \div 2 = 9{,}15 \text{ lbs.}$$

It has been found by experiment that for a beam *fixed at ends and loaded in the middle*,*

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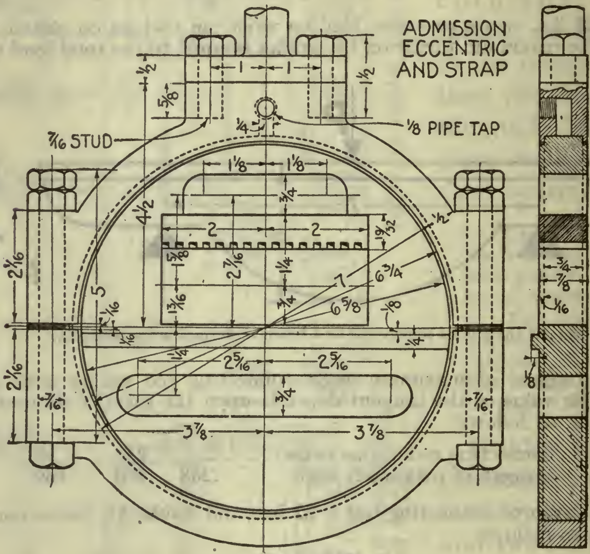
*NOTE.—Strictly speaking, the beam is ***semi-fixed*** because of the bending action of the bolts; however it is sufficiently accurate to assume the beam as fixed.

$$P=\frac{4}{3}\,\frac{R\,bd^2}{l}\ldots\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

in which P = breaking load in lbs.; R = modulus of rupture, or stress per sq. in. of extreme fibre; l = length, b = breadth, d = depth, in ins.; For cast iron, the value of R varies greatly according to quality. Thurston found 45,740 to 67,980 in No. 2 and No. 4 cast iron respectively.

Now, considering the guide as a beam, its length between supports (as in fig. 3,265) is 7 ins., depth, 2½ ins. and breadth ⅜ in. Substituting these in (1) and taking R at 50,000



FIGS. 3,266 and 3,267.—Admission eccentric and strap showing hardened 45° tooth shifter block at top of slot.

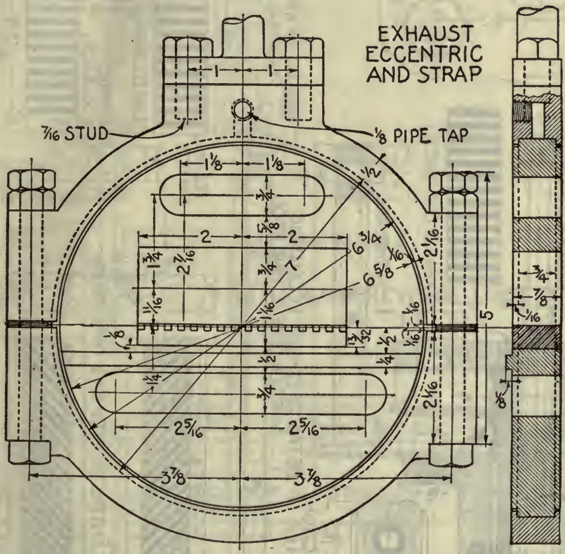
$$P=\frac{4}{3}\times\frac{60{,}000\times\frac{3}{8}\times(2\frac{1}{2})^2}{7}=26{,}786$$

and

$$\text{factor of safety}=\frac{\text{breaking load}}{\text{thrust per guide}}=\frac{26{,}786}{915}=29$$

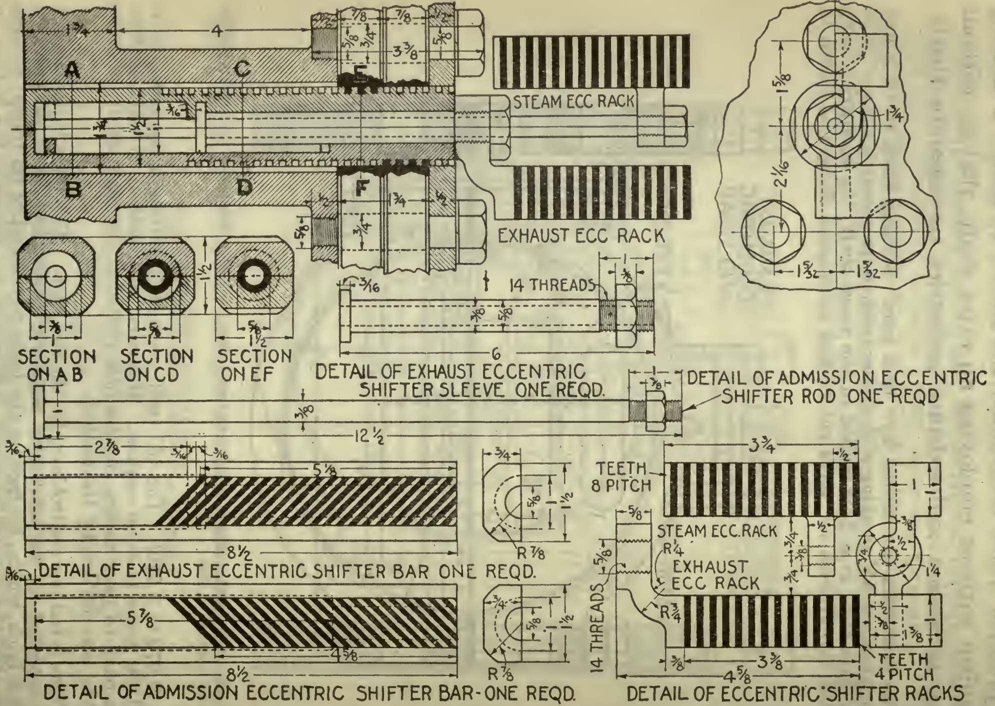
Variable Cut Off and Reverse Gear.—For obtaining variable cut off and reverse, shifting eccentrics are used, and it was some problem which confronted the author to design a shifting mechanism without employing an extra shaft, that is, without having admission and exhaust eccentrics on separate shafts.

The eccentrics are shifted by "rack bars" working back to back and having teeth at 45° meshing with similar teeth in the eccentric slots. These rack bars slide in a hole drilled in the main shaft, the eccentrics being guided by grooves or ways cut in two companion flanges as shown in figs. 3,270 to 3,270*l*. The 45° teeth method of shifting is not new, but is used on some traction engines. However, the method of locating these bars back to back in one circular hole drilled in the main shaft, and the control was devised by the author.



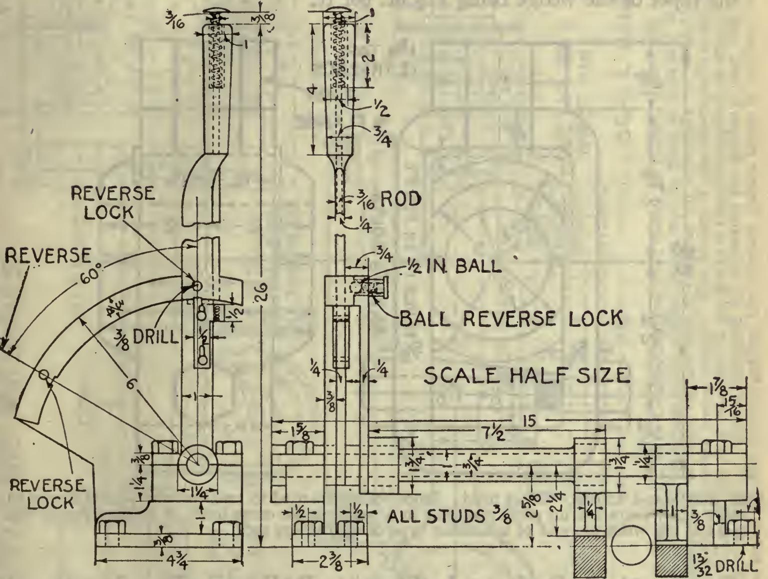
Figs. 3,268 and 3,269.—Exhaust eccentric and strap showing hardened 45° tooth shifter block at bottom of slot.

There is but one control lever for the two bars. This lever is permanently connected to the admission eccentric shifter arm, and is provided with a latch, which, by moving the lever to the end of its travel, will engage with the exhaust eccentric shifter arm by releasing the button on top of the lever. The exhaust gear automatically locks at each end of the


A
B
C
D
E
F
4
1¾
STEAM ECC RACK
EXHAUST ECC RACK
SECTION ON A B
SECTION ON CD
SECTION ON EF
14 THREADS
6
DETAIL OF EXHAUST ECCENTRIC SHIFTER SLEEVE ONE REQD.
DETAIL OF ADMISSION ECCENTRIC SHIFTER ROD ONE REQD
12½
2⅞
5⅛
8½
R ⅞
DETAIL OF EXHAUST ECCENTRIC SHIFTER BAR ONE REQD.
5⅞
4⅝
DETAIL OF ADMISSION ECCENTRIC SHIFTER BAR-ONE REQD.
TEETH 8 PITCH
STEAM ECC.RACK
EXHAUST ECC RACK
R¼
R¾
14 THREADS
3¾
3⅜
4⅝
TEETH 4 PITCH
DETAIL OF ECCENTRIC SHIFTER RACKS
1⅝
2⅟16
1⁵⁄₃₂

FIGS. 3,270 to 3,270*l*.—Eccentric shifting gear and details of parts. Fig. 3,270 and 3,270*a*, assembly; figs. 3,270*b* to 3,270*d*, sections of shifter bars, through AB, CD, and EF respectively; fig. 3,270*e* exhaust eccentric shifter sleeve; fig. 3,270*f*, admission eccentric shifter rod; figs. 3,270*g* and 3,270*h* admission eccentric shifter bar; figs. 3,270*i* and 3,270*j*, exhaust eccentric shifter bar; fig. 3,270*k*, admission eccentric rack; fig. 3,270*l*, exhaust eccentric rack.

quadrant by a ball lock (similar to that used in automobile transmissions) leaving the lever free to be placed in any cut up off position.. Thus, in operating, the control lever is moved to the end of the quadrant and button released which engages the exhaust arm. The lever is then moved to the other end of the quadrant where the exhaust arm engages with the ball lock. By pressing down the button on top the lever, it is released from the exhaust arm and is moved back to the desired cut off position and button released which locks lever in position.



FIGS. 3,270*m* and 3,270*n*.—Variable cut off and reverse lever control gear.

Cross Head.—This is a very simple affair consisting of a rectangular piece which carries the wrist pin bearing and provided with a neck to receive the piston rod as in figs. 3,270*p* and 3,270*o*.

Calculating for strength,

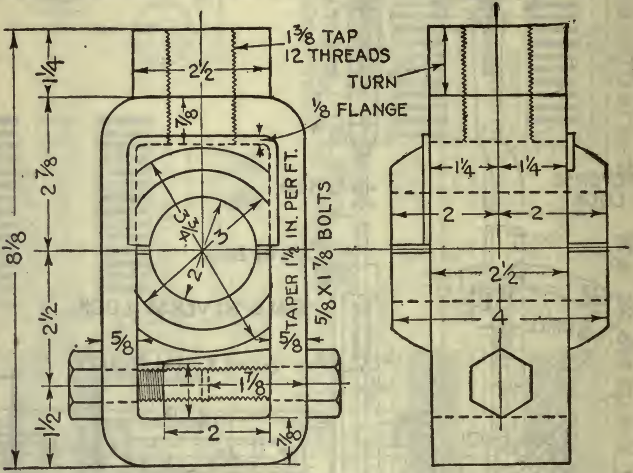
$$\text{area piston rod} = 1.49 \text{ sq. in.}$$

and making each arm strong enough to carry 7/10 of the load,

$$\text{area each arm} = 1.49 \times .7 = 1.04 \text{ sq. ins.}$$

Make arms, say 2×⅝ ins.

The ends are made somewhat thicker, and before drawing the cross head, the diameter of the wrist pin must be determined from the connecting rod design. The neck should be made long enough for ample adjustment of the piston rod. The wrist pin bearing has wedge adjustment, the taper of the wedge being 1½ in. per ft.



FIGS. 3,270*o* and 3,270*p*.—Cross head. Since rollers are used to transmit the lateral thrust to the guides, there are no gibs, resulting in a very simple cross head, it being virtually the same as the wrist pin end of a familiar type of connecting rod.

Connecting Rod and Thrust Rollers.—The length of connecting rod means the *distance between centers of wrist pin and crank pin bearings*. Experience has generally led to a

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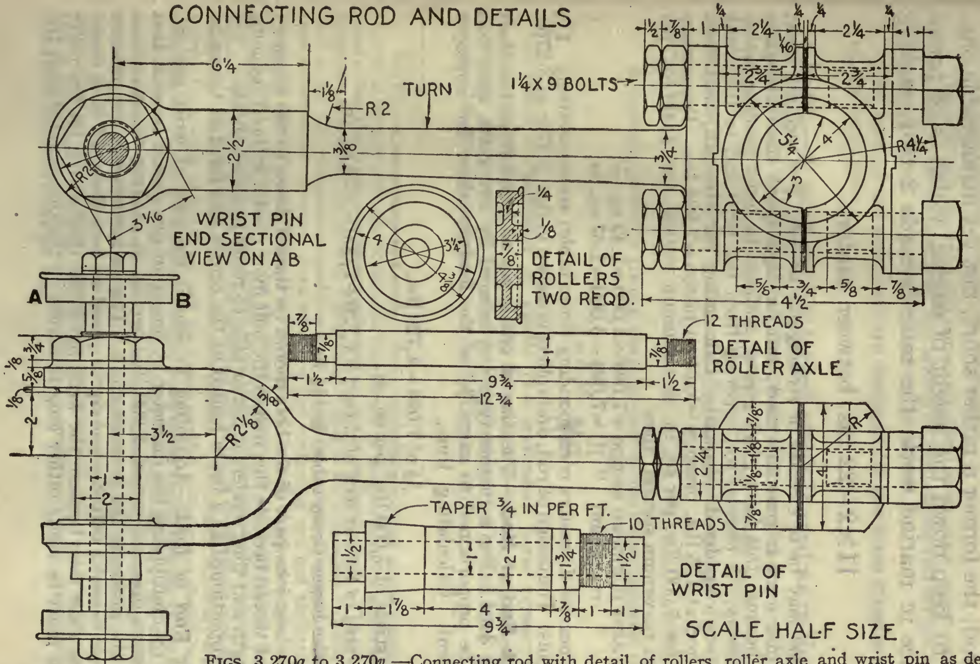
NOTE.—The area of the cross head slipper or gib (the surface on which the thrust is taken) should in no case be less than will admit of a pressure of 400 lbs. per sq. in. and for good working these surfaces which take the thrust when going ahead should be sufficiently large to prevent the maximum pressure exceeding 100 lbs. per sq. in. When the surfaces are kept well lubricated, these allowances may be exceeded.—*Seaton*.

NOTE.—Some British builders of portable engines restrict the pressure between the guides and crossheads to less than 40, sometimes 35 lbs. per sq. in.—*Kent*.



FIGS. 3,270*q* to 3,270*v*.—Connecting rod with detail of rollers, roller axle and wrist pin as described in the text. The rollers transmit the lateral thrust to the guides, being used in place of gibs and is one of the unique features of the engine. Gibs are objectionable not only on account of friction but especially on account of their tendency to heat—in fact on some large engines the guides are water cooled. The object of the rollers is to avoid both of these objectionable features. ***In operation*** the rollers touch only one pair of guides at a time (depending on whether the engine be in forward or reverse motion) hence they turn freely. This is obtained by adjusting the guides to say .002 in. clearance or about the thickness of a piece of tissue paper for the reverse guides, the forward guides being in contact with the rollers.

ratio of length of connecting rod to length of stroke of 2 or 2½ to 1, the latter giving, according to Thurston, a long and easy working rod, the former a rather short, but yet manageable one. In order to keep down the height of engine, which is of prime importance in marine work, the ratio is made 2 to 1, making the rod

$$11 \times 2 = 22 \text{ ins. between centers.}$$

The rod may be of circular or rectangular section. For medium speed the circular form is generally used, and for high speed the rectangular section, because this form gives greater stiffness to resist bending action due to inertia. A circular rod tapering from the wrist end to the crank end is here considered adequate. Calculating for strength, maximum load on rod at 125 lbs. pressure = load on piston = 7,094 lbs. Proportioning rod for 5,000 lbs. per sq. in. at minimum section, and 3,000 lbs. per sq. in. at maximum section, then

Area minimum section = 7,094 ÷ 5,000 = 1.42 sq. in.
" maximum " = 7,094 ÷ 3,000 = 2.37 sq. in.

and the nearest diameters corresponding (from table) are respectively 1⅜ and 1¼ ins. The design of rod, and rollers is shown in figs. 3,270*r* to 3,270*q*.

Crank End of Rod.—The size of crank pin as found from the shaft design is 3×4. For a marine type end secured by two bolts, each bolt should be proportioned to carry ⅔ of the load to allow for improper adjustment. Calculating thus,

⅔ of 7,094 = 4,729 lbs.

For a stress of 7,000 lbs. per sq. in.

area of bolt at root of thread = 4,729 ÷ 7,000 = .676 sq. in.

nearest size larger (from table) is 1¼ in.

On the best jobs part of the bolt should be turned down to the diameter at root of thread because this makes the bolt more elastic without reducing its strength and thus renders it better able to resist repeated shocks. The cap is proportioned as a beam loaded at the center and fixed at the ends.

Wrist End of Rod.—According to Seaton the area of wrist pin calculated by multiplying the diameter by length of pin should be such that the pressure does not exceed 1,200 lbs. per sq. in., taking the maximum load on the piston as the pressure on it. The author believes the pressure should not exceed 1,000 lbs. per sq. in., and as here designed it is somewhat less than this, because the wrist pin forms the bearing for the roller axle and must be made rather long to divide up the distance between the rollers.

The design begins with the rollers, the diameter of which must correspond with the distance between the guides (obtained from fig. 3,259). The wrist pin is bored to receive the roller axle the latter being fast to the rollers and turning within the wrist pin.

The wrist pin fits into the connecting rod jaws on a taper and is secured by a nut. On each side of the jaws a section of the rod serves as a bearing for the feed and air pump drive rods, as shown.

The jaws of the wrist end are designed for the same stresses as the arms of the wrist pin.

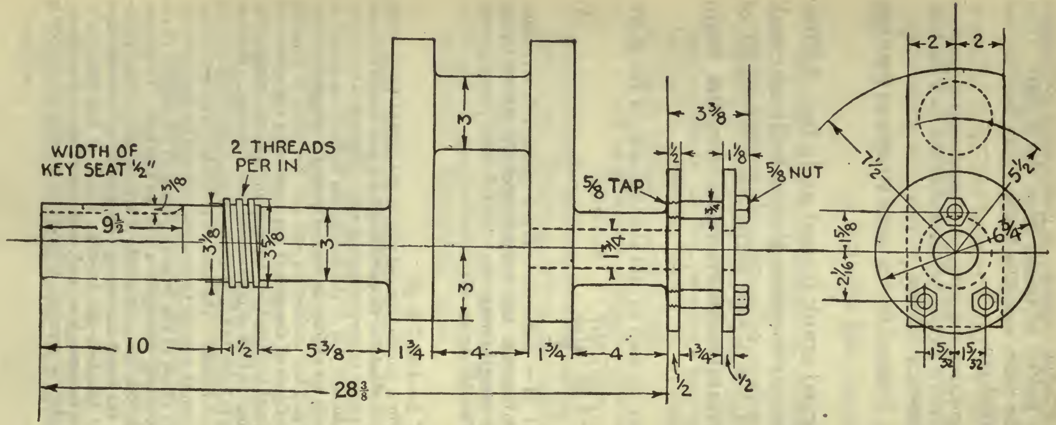
Shaft.—A solid double forged crank is used as in figs. 3,270*w* and 3,270*x*. While this is more expensive than a built up crank it makes an elegant job and the crank arms may be made of less breadth which make available more room for the main bearings.

A formula given by Unwin may be used to find the diameter of the shaft:

$$d = 4.55 \sqrt[3]{\frac{H}{N}} \qquad (1)$$

in which d = diameter of shaft; H = indicated horse power; N = revolutions per minute. In using the formula the horse power should be taken at, say .50 instead of 30, as by lengthening the cut off the horse power may be greatly increased. Substituting in (1)

NOTE.—***Connecting rod ends.*** For a connecting rod end of the marine type, where the end is secured with two bolts, each bolt should be proportioned for a safe tensile strength equal to two-thirds of the maximum pull or thrust in the connecting rod. The cap is to be proportioned as a beam loaded with the maximum pull of the connecting rod and supported at both ends. The calculation should be made for rigidity as well as strength, allowing a maximum deflection of .01 inch. For a strap and key connecting rod end the strap is designed for tensile strength, considering that two-thirds of the pull of the connecting rod may come on one arm. At the point where the metal is slotted for the key and gib, the straps must be thickened to make the cross section equal to that of the remainder of the strap. Between the end of the strap and the slot, the strap is liable to fail in double shear, and sufficient metal must be provided at the end to prevent such failure. The breadth of the key is generally one-fourth of the width of the strap, and the length parallel to the strap, should be such that the cross section will have a shearing strength equal to the tensile strength of the section of the strap. The taper of the key is generally about five-eighth's inch to the foot. In modern high speed engines it is customary to make the connecting rods of rectangular instead of circular section, besides being parallel, and the depth increasing regularly from the cross head end to the crank pin end. According to Grashof, the bending action of the rod due to its inertia is greatest at .6 the length from the cross head end, and, according to this theory, that is the pioint at which the section should be greatest, although in practice the section is made greatest at the crank pin end.



FIGS. 3,270*w* and 3,270*x*.—Crank shaft and eccentric guides. The forward portion is bored for the eccentric shifter bars. On the rear end a coarse plus thread is cut for the roller thrust bearing adjusting collar.

$$d = 4.55 \sqrt[3]{\frac{50}{250}} = 2.6, \text{ say } 2\tfrac{1}{2} \text{ ins.}$$

Now, the front section of the shaft must be bored to receive the valve gear shifter bars, hence the diameter must be increased, thus:

area 2½ shaft = 4.91 sq. in.
" 1¾ bore = 2.41 " "

area solid portion of shaft 7.32 sq. in. corresponding to 3 1/16 shaft diameter—make shaft 3 in. diameter. The crank pin is usually made the same diameter as the shaft and proportioned for a load of 500 lbs. per sq. in. of projected area.[*]

For a 3×4 pin, the projected area is $3\times4=12$ sq. ins. and

$$\begin{aligned}\text{load} &= \text{load on piston} \div \text{projected area}\\ &= 7{,}094 \div 12 = 591 \text{ lbs. per sq. in.}\end{aligned}$$

being 91 lbs. in excess of the usual allowance, but is permissible in this case because the normal working load on engine is considerably less than here taken.[†]

According to Unwin the thickness of crank may be from .6 to $.8d$, and its breadth from 1.05 to $1.3d$. Accordingly

$$\begin{aligned}.6\times3&=1.8, \text{ say } 1\tfrac{3}{4} \text{ ins.}\\ 1.3\times3&=3.9, \text{ say } 4 \text{ ins.}\end{aligned}$$

The forward end of the shaft is provided with a flange, which carries a similar flange attached by three bolts, the object of these flanges being to hold in place the two shifting eccentrics. The grooves of these flanges register with tongues integral with the eccentric forming guides in the direction of shift and together with the bolts preventing any angular motion between eccentrics and flanges.

[§]**Feed Pump.**—An engine driven feed pump is a very desirable device for feeding the boiler because of 1, its compactness,

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[§] NOTE.—The author does not believe in independent pumps (either feed or air) especially on small or medium size vessels; because of 1, extra space, 2, extra weight, 3, unsatisfactory working, 4, poor efficiency, of the independent pump, and 5, because two pipe lines are better than four, all of which leave little or no room for argument in favor of independent pumps.

[*] NOTE.—***Projected area*** means the area obtained by multiplying length by diameter of pin.

[†] NOTE.—The designer should aim at obtaining simple dimensions; it would indeed be ridiculous to make the pin say $3\frac{1}{16}\times4\frac{1}{32}$ to proportion it exactly to a given allowable working pressure, there is more chance of making mistakes both in making the drawings and in building the engine.

NOTE.—The empirical proportions adopted by builders for the crank arm will commonly be found to fall well within the calculated safe margin. These proportions are, from the practice of successful designers about as follows: ***For the wrought iron crank,*** the hub is 1.75 to 1.8 times the least diameter of the inserted portion of the pin, and the depths are, for the hub 1 to 1.2 the diam. of shaft, and for the eye, 1.25 to 1.5 the diam. of pin. The web is made .7 to .75 the width of adjacent hub or eye, and is given a depth of .5 to .6 that of adjacent hub or eye. ***For cast iron crank***, the hub and eye are a little larger, ranging in diameter respectively from 1.8 to .2 and from 2 to 2.2 times the diameter of shaft and pin. The flanges are made at each end of nearly the full depth of hub or eye. Cast iron has, however, fallen very generally into disuse.—*Thurston.*

2, light weight, 3, minimum amount of piping, 4, automatic speed control, etc. The type here selected is the single acting plunger driven from the cross head through the familiar walking beam transmission.

To calculate the size required, make capacity sufficient to pump *twice the feed water and allow say 5% for slip*, that is

$$\text{*Capacity of pump} = 2(20 \times 30) + 5\% = 1{,}260 \text{ lbs. per hour}$$

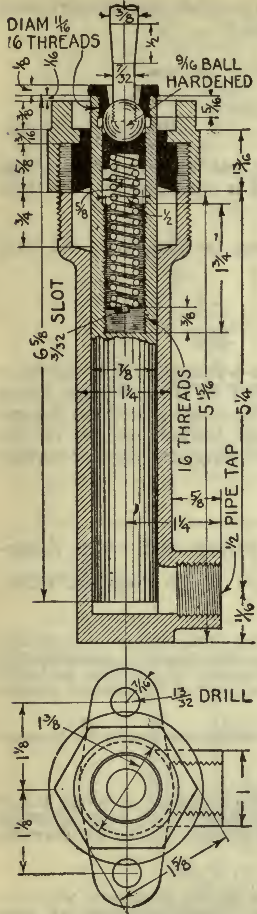
$$\text{water per minute} = 1{,}260 \div 60 = 21 \text{ lbs.}$$

The pump being single acting makes one delivery stroke per revolution of the engine and at 250 rev. per minute, hence

$$\text{water per delivery stroke} = 21 \div 250 = .084 \text{ lb.}$$

Assuming a hot well temperature of 120° Fahr., 1 cu. ft. of water @ 120° weighs 61.74 lbs., hence 1 lb. of water = 1,728 ÷ 61.74 = 27.99 cu. in., and on this basis

$$\text{pump displacement per delivery stroke} = 27.99 \times .084 = 2.35 \text{ cu. in.}$$



FIGS. 3,270*y* and 3,270*z*.—Feed pump. Note the absence of air chamber. The author, who had several years' experience designing large pumps for city water works found that the so-called air chambers were virtually water chambers, because unless means were provided to keep them filled with air, they soon became filled with water and ceased to perform their function of cushioning the shocks. Accordingly such device is here dispensed with, and a spring cushion provided which is expected to absorb the shocks even at high speed. No valves are built into the pump casting, a tee fitting and two check valves being used, connected with close nipples.

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*NOTE.—The factor 20 is the "expected" consumption of feed water per hour, and 30 the indicated horsepower.

Evidently various values may be given to stroke and diameter of plunger to obtain the required displacement, and to properly choose the best ratio, a table should be made (by aid of slide rule) thus

Feed Pump Sizes

(for 2.35 cu. in. displacement per delivery stroke)

Diameter..............	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{13}{16}$	1
Stroke................	5.35	3.92	4.54	3

The next step is to make a center line "layout" of the driving gear, and in doing this the stroke of pump will depend somewhat on the distance between the engine columns and cylinder axis and space available on bed plate for the pump, all of which must be considered in fixing the stroke. The size of pump as finally determined is $\frac{7}{8} \times 4\frac{1}{2}$, the stroke being a little over a half inch in excess of that given in the table to allow for lost motion due to the spring cushion devise used instead of an air chamber.

The size of pipe inlet to pump is based on a flow of 250 ft. per minute, thus

$$\text{*area of inlet} = \frac{21 \times 27.99}{250 \times 12} = .196 \text{ sq. in.}$$

Nearest (larger) size pipe (from table) is ½ in. (area .304 sq. in.). The delivery pipe should be made this size, and the suction one size larger, or ¾ in. Figs. 3,270*y* and 3,270*z* show the feed pump design.

Feed Pump Drive.—This, as shown on figs. 3,271*a* to 3,271*f*, consists of 1, connecting rod with spring cushion ball joint on plunger, 2, walking beam, and 3, drive rod. In proportioning the parts, first calculate the load on the plunger, at 125 lbs. pressure thus:

area of $\frac{7}{8}$ circle from table = .601 sq. in.
load on plunger = $.601 \times 125 = 75$ lbs.

For a load of 5,000 lbs. per sq. in.

area connecting rod at root of thread = $75 \div 5{,}000 = .015$ sq. in.

Nearest larger size (from table) is ¼ in. (area .027)—make diameter at ball $\frac{7}{32}$ in., and for rigidity increase diameter of body of rod to say ⅜ in.

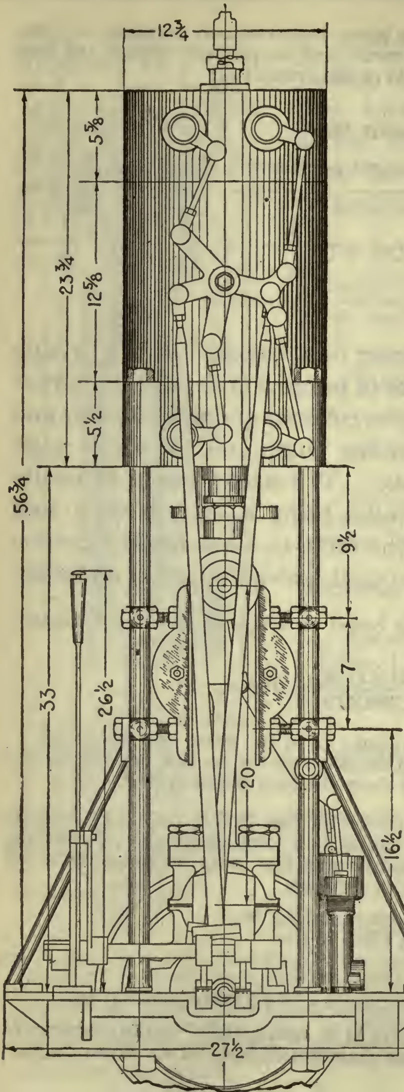


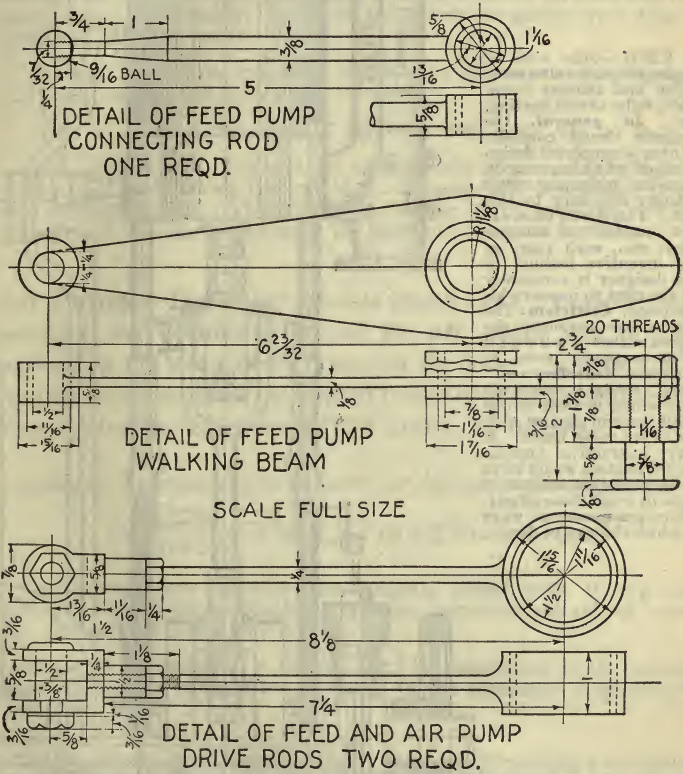
FIG. 3,271.—Front view of engine showing valve gear roller cross head, pumps, etc. ***The general proportions*** of the engine are here clearly seen. Its unusual height is due to the long stroke employed to increase the effectiveness of the jacket. The extra width of the bed plate, however, gives a footing for diagonal braces which make the frame amply stiff. Of course, in design a less extreme bore-stroke ratio may be employed. ***It should be noted*** that the roller cross head permits of a very short connecting rod (1.81 to 1), thus leaving room for a longer stroke than would otherwise be possible without increasing the height. The long stroke is used, because, as mentioned, it is *more efficient than a short stroke.* The reason for this, as explained in the discussion of the steam jacket (pages 1,655 to 1,657) is because the ***ratio*** *of the jacketted surfaces to the cylinder volume is* ***greater*** *with the long stroke than with the short stroke.* ***Accessibility.*** This is an important consideration especially with engines of small or medium sizes. The designer should aim to so arrange all parts that have to be adjusted or oiled that they will be easy to reach, otherwise they will not receive proper attention. The frame here used is the most "open" type, and as seen in this view and in fig. 3,271*j*, the parts can be easily adjusted with ordinary wrenches. ***In general,*** *avoid designs which require special wrenches.*

Consider the longer arm of the walking beam as a beam fixed at one end and loaded at the other, then using the formula

$$P = \frac{1}{6}\,\frac{R\,b\,d^2}{l} \quad \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

and taking $R = 100{,}000$ for steel forging, $b = \frac{1}{8}$, d (effective) $= 1$ and $l = 6\frac{23}{32}$, and substituting in (1)

$$P = \frac{1}{6} \times \frac{100{,}000 \times \frac{1}{8} \times 1^2}{6.72} = 310 \text{ lbs.} \quad \ldots\ldots\ldots\ldots\ldots (2)$$



Figs. 3,271*a* to 3,271*f*.—Feed pump drive, consisting of connecting rod, walking beam and drive rod. The same design drive rod is used for both feed and air pumps.

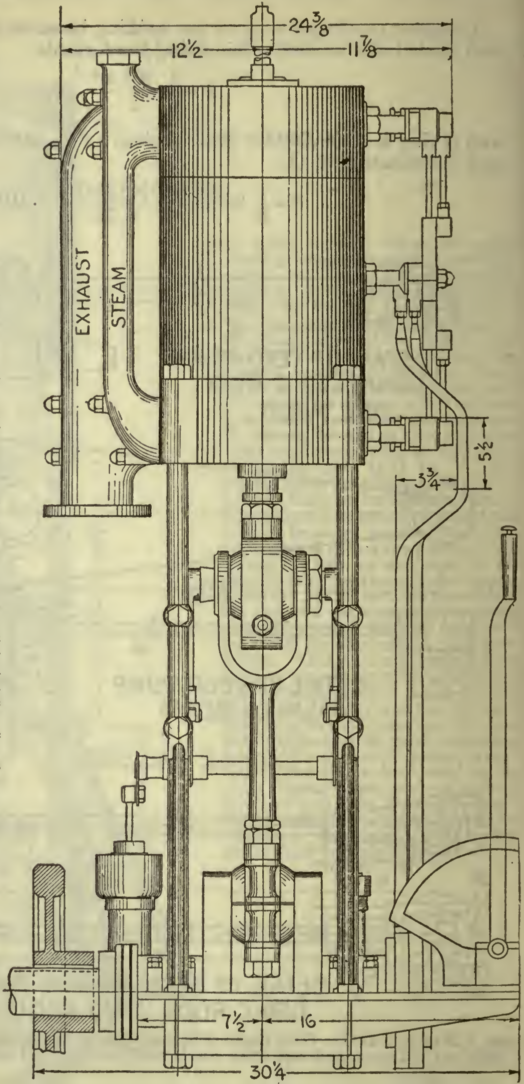


FIG. 3,271*j*.—Side view of engine showing valve gear, steam and exhaust manifolds, roller thrust bearing, etc. ***In general,*** the designer should carefully go over a completed design in an effort to improve on same and make any minor changes necessary to this end. This is why there are new "models" of automobiles, etc., each year, as the inventive instinct of the designer is constantly on the alert to improve his creations. ***Criticism.*** The author has *purposely* designed rather awkward *offset* eccentric rods to illustrate ***"interference."*** Here straight eccentric rods would in their movement come in contact with the lower valve heads. A very instructive exercise for the student would be to re-design the eccentric drive so as to avoid these offsets. There are numerous ways in which this can be done.

Now the ratio of walking beam arms is

$$6.72 \div 2.75 = 2.44$$

from which the load at the loaded end is

$$75 \div 2.44 = 30.7 \text{ lbs.}$$

which divided by the result in (2) gives a factor of safety of

$$310 \div 30.7 = 10$$

but the metal at the fixed end, being at some distance on each side of the neutral axis considerably increases the factor of safety over that just obtained, which is ample.

The load carried on the drive rod is 30.7 lbs. when pumping against 125 lbs. pressure, and for a 5,000 lb. stress,

area at root of thread $= 30.7 \div 5{,}000 = .006$, say, .01 sq. in.

Now, area at root of thread for smallest size given in table (¼ in.) is .027 sq. in. and making the rod ¼ in. diameter will give ample strength and stiffness besides. The very large bearing at the wrist pin end of the drive rod is necessary because of the large pin diameter required.

Air Pump.—The object of this pump is to remove from the condenser the ***condensate*** *and the* ***air*** *which was originally contained in the water when it entered the boiler.* Hence the size of the air pump must be calculated from these conditions, making allowance for the efficiency of the pump, as follows:

*Volume of condensate to be removed per delivery stroke of pump is

$$2 \times \frac{20 \times 30}{60 \times 250} = .08 \text{ lb., or } .08 \times 27.99 = 2.24 \text{ cu. in.}$$

Oridinarily water contains, mechanically mixed with it, $^1/_{20}$ of its volume of air under atmospheric pressure, hence actual vol. of water to be pumped $= {}^{19}/_{20}$ of $2.24 = 2.13$ cu. in.

and if a 28 in. vacuum (=.943 lb. abs.) be maintained in the condenser, then the volume of air *after expanding* in the condenser (neglecting temperature) is

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*NOTE.—20 =lbs. steam per hour per *i.h.p.*; 30 = *i.h.p.*; 60 =minutes in one hour; 250 = *r.p.m.*; multiplying the fraction by 2 gives condensate *per stroke* instead of *per revolution;* because the pump is *single acting;* 27.99 = volume in cu. in. of lbs. of condensate.

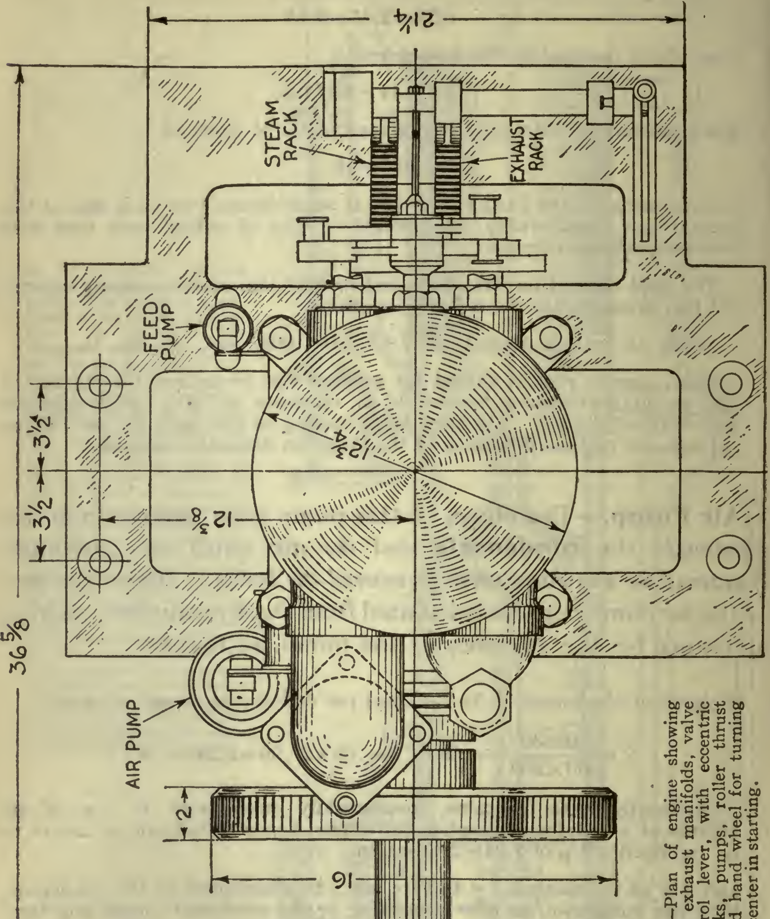


Fig. 3,271*h*.—Plan of engine showing steam and exhaust manifolds, valve gear, control lever, with eccentric shifter racks, pumps, roller thrust bearing and hand wheel for turning engine off center in starting.

$$(\tfrac{1}{20}\text{ of }2.13)\times\frac{14.7}{.943}=1.66\text{ cu. in.}$$

Now, if the temperature of the condenser be taken at 120°, and the initial temperature of the feed water at 70, then

$$\text{quantity of air}=1.66\times\frac{120+461}{70+461}=1.82$$

and the total volume to be pumped *per delivery stroke* is

$$2.13+1.82=3.95\text{ cu.in.}$$

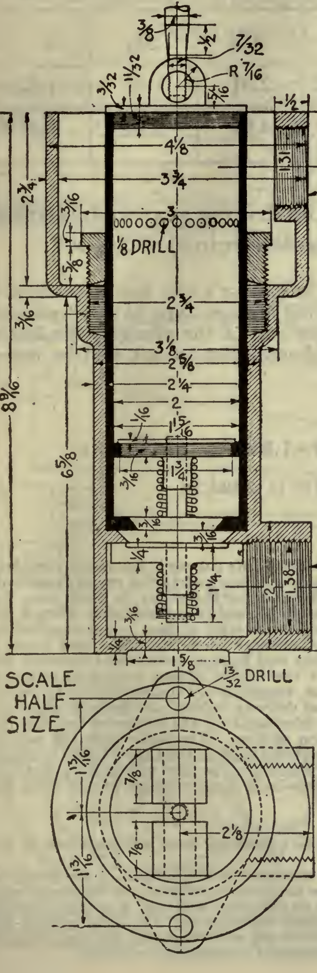
and taking the efficiency of the pump at 35%, then

$$\text{displacement of pump}=3.95\times.35$$
$$=11.3\text{ cu. ins.}$$

Similarly as was done in the feed pump calculation make a table (using slide rule) of various sizes having the displacement just found, thus:

NOTE.—In designing an air pump, it should be remembered that *the work done depends upon the* ***volume*** *of air and water delivered against atmospheric pressure, and* ***not*** *upon the* ***size*** *of the pump*. It is accordingly not objectionable if the pump be a little over the theoretical size.

NOTE.—In designing a trunk piston single acting pump having foot valves, it should be noted that the pump is in a sense double acting. Thus on the down stroke a volume is discharged equal to the sectional area of trunk multiplied by the stroke.



FIGS. 3,271*i* and 3,271*j*.—***"Zero clearance"*** air pump, adapted to installations where the hot well inlet is below the level of the air pump discharge. The bucket valve is always under a head of water, hence is water sealed at all times. The arrangement of the valves permits of practically zero clearance by close adjustment, hence all the air is forced out of the barrel on each down stroke, thus tending to produce the maximum vacuum. ***In operation,*** the condensate oozes out of the multiplicity of small holes drilled in the top of the hollow plunger and flows to hot well by gravity. The stuffing box is water sealed.

Air Pump Sizes

(for 11.3 cu. in. displacement per delivery stroke)

Diameter..........	2	2¼	2½	2¾	3
Stroke............	3.6	2.85	2.31	1.91	1.62
Area plunger......	3.14	3.98	4.91	5.94	7.1

Now make a center line layout of the driving gear to determine best size of pump. The size as thus determined is 2¼×3.

In the design figs. 3,271*i* and 3,271*j*, the author's aim was to produce a "zero clearance" pump of the single acting plunger type having foot and bucket valves and an outlet at the upper end of the plunger, the latter feature making it necessary that the discharge inlet to hot well be lower than the top of the pump barrel.

The valves are calculated thus:

$$\text{plunger speed} = (3 \times 2) \times 250 = 1{,}500 \text{ ins. per minute}$$

and a flow of 400 ft. per minute past valve is equal to

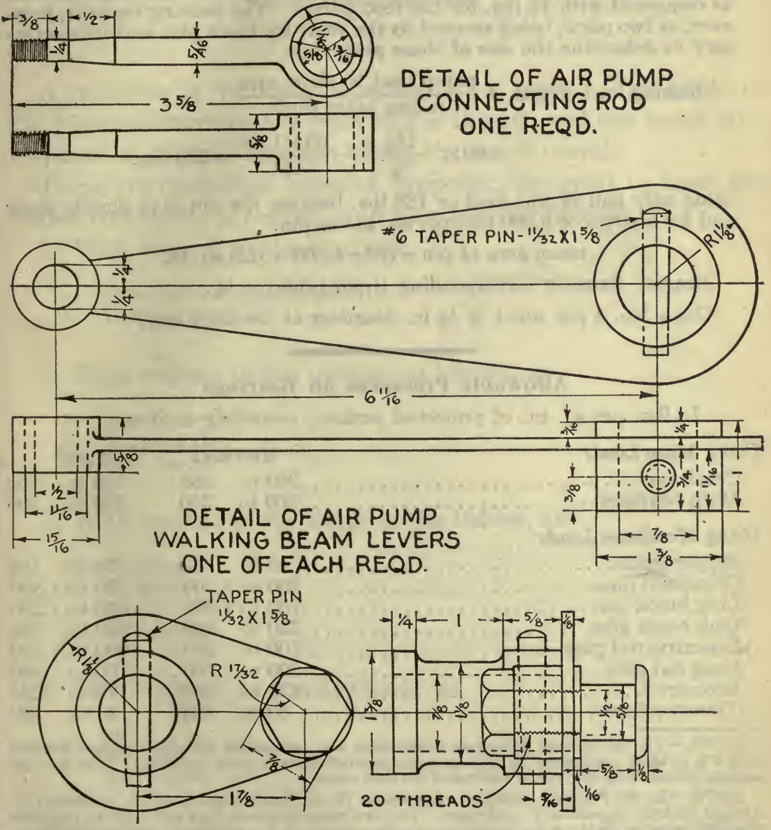
NOTE.—***The efficiency of the single acting vertical air pump*** is due to the certainty of its action in taking the water, etc., through the bucket valves, and to the valves from their position so readily closing when required, there is also time for the water to drain into the bottom of the pump on its upstroke and it is collected there ready for the bucket when it descends. The flow is always in one direction, so that the velocity of flow is not checked by diversion. The water always lies on the valves so as to render them air tight, and there is very little clearance space, as a rule, between the foot and bucket valves, and between the bucket and head valves. The want of efficiency of the double acting horizontal pump is caused by the reverse of some of the above conditions, especially by the failure of the valves in closing, and to the large space between the foot and delivery valves, also by leakage at the gland of the rod and past the bucket which is only lubricated by the water on the bottom, and in no small degree by the ever changing direction of flow. The latter defect is proved by stopping one end of the pump, when it is often found (especially in the case of badly designed pumps, etc.) that the vacuum is not very materially altered. The foot valves are sometimes kept covered with water by allowing the water to pass back again through a pipe from the hot well to the pump chamber.—*Seaton.*

NOTE.—The capacity of the air pump should be calculated from consideration of the conditions under which it is to work, ... and suited to practice by an allowance made for the efficiency of the pump. If the pump be single acting and well designed, and is working under favorable conditions, its efficiency may be taken at .6; and if the reverse of this, .4; generally its efficiency is about .5; so that the size in practice should be double that given by theoretical calculation. The efficiency of the double acting pump varies from .5 to .3, and generally is not more than .35; the size for good working should be nearly three times that of the theoretical calculations.—*Seaton.*

$$\frac{400\times 12}{250\times 2}=9.6 \text{ ins. per stroke}$$

and the opening to give this flow, or

$$\text{valve area}=11.3\div 9.6=1.18 \text{ sq. ins.}$$



FIGS. 3,271*k* to 3,271*p*.—Air pump drive. Figs. 3,271*k* and 3,271*l*, connecting rod; figs. 3,271*m* and 3,271*n*, long arm of walking beam; figs. 3,271*p* and 3,271*q*, short arm of walking beam. The connecting rod is the same design as used for the feed pump (see page 1,727.)

a 1½″ (flat) valve opening ¼ in. will give this area, but to obtain zero clearance use 45° bevel valve and make as large as possible. The air pump design is shown in figs. 3,271*i* and 3,271*j*.

Air Pump Drive.—This is very similar to the feed pump drive, and is of the same proportions as the loads are very nearly the same, thus, area plunger $= 3.98$ sq. ins. and for 15 lbs. pressure, load $= 3.98 \times 15 = 60$ lbs. as compared with 75 lbs. for the feed pump. The walking beam is, however, in two parts, being secured to the shaft by taper pins and it is necessary to determine the size of these pins, thus

$$\text{shearing load on pin} = \frac{\text{pump load} \times \text{short arm}}{\text{radius beam shaft}}$$

$$= 60 \times \frac{1\frac{7}{8}}{\frac{7}{16}} = \frac{60 \times 1.88}{.44} = 256 \text{ lbs.}$$

using only half of this load or 128 lbs. because the pin is in double shear and for a stress of 5,000 lbs. per sq. in. on pin.

$$\text{mean area of pin} = 128 \div 5{,}000 = .026 \text{ sq. in.}$$

Nearest diameter corresponding (from table) is ³⁄₁₆.

Use a No. 4 pin which is ¼ in. diameter at the large end.

Allowable Pressures on Bearings

In lbs. per sq. in. of projected surface, according to *Bragg*,

	Merchant	*Naval*
Using Mean Loads		
Crank pin	200 to 250	250 to 300
Main bearings	200 to 350	250 to 500
Using Maximum Loads		
Slipper guide	60 to 80	70 to 100
Crosshead pins	850 to 1,200	1,200 to 1,800
Link block pin	750 to 1,000	850 to 1,200
Link block gibs	250 to 400	350 to 500
Eccentric rod pins	700 to 950	900 to 1,100
Drag rod pins	500 to 700	700 to 800
Eccentrics	150 to 200	175 to 225
Thrust collars	50 to 80	80 to 100

NOTE.—The air leakage in surface condensers is an extremely variable quantity ranging from 1% in tight condensers to 25% or more in condensers in poor condition. The average leakage is from 5 to 10% of the volume of the feed water.

NOTE.—In the ***Parsons vacuum augmenter*** the air is exhausted from the condenser by a steam jet into an auxiliary condenser. The dry vacuum pump does not have to maintain the high vacuum in the main condenser, as the vacuum in the auxiliary condenser is lower, hence a smaller air pump may be used. ***Steam consumption*** is usually from .6 to 1½% of that of the main turbine. The net saving in the average condenser due to augmenter is about 5%. With tight condensers the saving may be negligible.

ADDENDA TO CHAPTER 52

PROPELLERS

A propeller is virtually *a section of a screw.* The angle of the blades determine the *pitch* of the thread, and the water corresponds to the nut through which the screw travels.

There are numerous types of propeller, designed to meet the varied conditions of service, and they may be classed:

1. With respect to the pitch as
    - *a.* True screw (constant pitch)
    - *b.* Variable pitch
2. With respect to the number of blades, as
    - *a.* Two blade
    - *b.* Three blade
    - *c.* Four blade
3. With respect to the shape of the blades, as
    - *a.* Weedless
    - *b.* Elliptical
    - *c.* Round
    - *d.* Straight tip
4. With respect to construction, as
    - *a.* Solid
    - *b.* Built up
    - *c.* Reversible

Slip.—In the case of a screw advancing through a nut, each revolution will produce a forward travel equal to the pitch of the screw. With a propeller, as before stated, the water is the nut and the *pitch* is the distance it would advance each revolution, if there were no slip. In practice there is always more or less slip.

There are two kinds of slip.

1. Real.
2. Apparent.

The ***real slip*** *is the velocity (relative to water at rest) of the water projected sternward.*

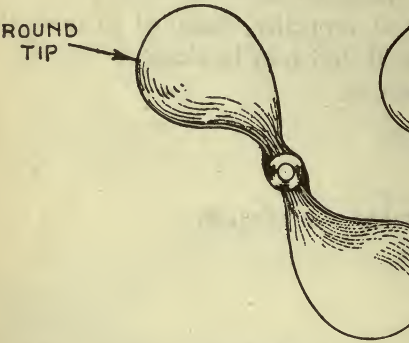


FIG. 3,271*q*.—Two blade round tip propeller.

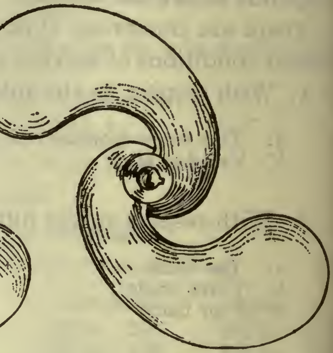


FIG. 3,271*r*.—Two blade round tip weedless propeller. This type propeller will clear itself of sea weed or other marine growths while running and should be installed as close to the strut or stern bearing as possible to prevent weeds gathering on the propeller shaft. The rudder should be tapered off on its forward edge below the propeller shaft so that the weeds will pass under the rudder.

The ***apparent slip*** *is* **the difference** *between the speed of the boat and the speed of the propeller*, that is, the product of the pitch of the propeller by the number of revolutions.

Thus, say a 20×30 propeller (20 in. diameter with a pitch of 30 ins.) would theoretically travel with the boat 30 ins. per revolution. At 500 *r.p.m.* it would travel in one minute $30 \times 500 \div 12 = 1{,}250$ feet, or in one hour $1{,}250 \times 60 \div 5{,}280 = 14.2$ miles—this is the theoretical speed of the propeller.

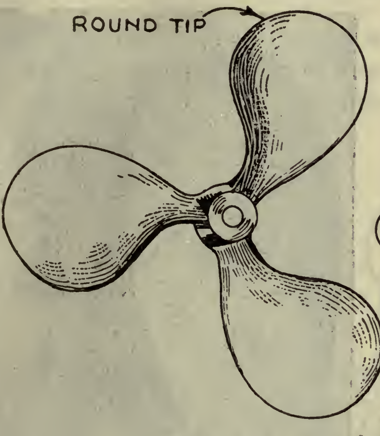
It should be carefully noted that when the word ***slip*** *is used unqualified it means the* ***apparent slip.***

Now in the above example if the slip be say 10% then the distance travelled by the boat would be

$$14.2 \times .9 = 12.8 \text{ mile per hour,}$$

thus, there is a loss of $14.2 - 12.8 = 1.4$ miles per hour due to the 10% slip.

The apparent slip or "slip" is sometimes negative, due to the working of the propeller in disturbed water which has a forward velocity, following the ship. Negative apparent slip is an indication that the propeller is not suited to the lines of the boat, or to a lack of *live* water for the propeller to work in.



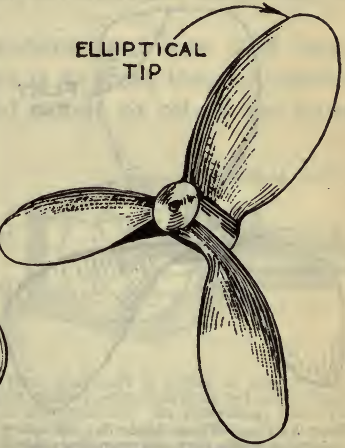


FIG. 3,271*s*.—Three blade round tip propeller. Some authorities maintain that the most important working surface of a propeller is located about ⅔ of the distance out from the hub, and that the greatest driving effort is exerted from that point to the end of the blade; also that the portion toward the hub develops comparatively little driving power and should therefore be designed merely for strength and to offer the least stream line resistance.

FIG. 3,271*t*.—Three blade elliptical tip propeller. The elliptical shape is favored by some because the blades are of simple outline, the contour offering a smooth curve at every point, without sharp angles, and permitting the advancing edge to offer minimum resistance in cutting the water and the leaving edge to allow the water to close over it with minimum chance of *cavitation*.

Normal and Abnormal Slip.—The percentage of slip varies considerably. Assuming the propeller used is correct for the installation, normal slip, as given by Hyde Windlass Co., is "from 15 to 25% depending upon the type of boat. In some cases, however, if the hull of the boat be of heavy construction and of bluff lines, a slip of 30% is common." Evidently this refers to gas engine installations, because gas engines usually run at speeds too high for good propeller efficiency.

Seaton states that "the slip generally should be about 8 to 10% at full speed in well formed vessels with moderately fine lines; in bluff cargo boats, it rarely exceeds 5%"—this relates to steam rigs, and the wide difference between 30 and 5% slip may be attributed to the better combination of engine speed and propeller dimensions in the case of the steam engine.

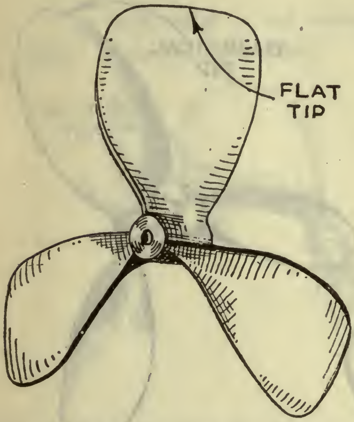


FIG. 3,271*u*.—Three blade flat tip heavy duty propeller, for boats operating against strong tides, currents or in rough weather against strong head winds, or for towing.

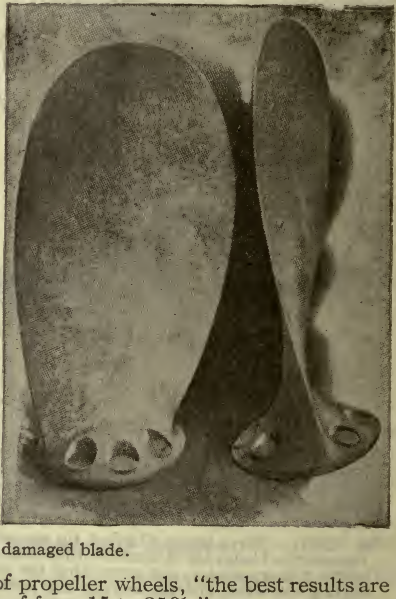


FIG. 3,271*v*.—Two views of blade of a built up propeller. This construction is desirable for large propellers to reduce size of castings and permit replacement of a damaged blade.

According to one manufacturer of propeller wheels, "the best results are obtained on motor boats with a slip of from 15 to 25%."

Pitch Ratio.—By definition, the pitch ratio is *the ratio of the pitch of the propeller to its diameter.*

*Thus in a 20×30 propeller the pitch ratio = 30 ÷ 20 = 1.5

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*NOTE.—It should be remembered in stating the dimensions of a propeller that the diameter is given first and then the pitch. Thus a 20×30 propeller means one of 20 ins. diameter and 30 ins. pitch.

Light hulls which drive easily may be fitted with propellers of reasonably high pitch, while hydroplanes and extremely light racing boats frequently use ratios as high as 1.75 and 2.

In general practice a ratio higher than 1.5 will produce abnomal slip and should be avoided. A safe rule is ***the heavier the hull, the lower the*** *pitch ratio*.

Installations have been made on extremely heavy scows and canal boats where a ratio as low as .45 has been used and satisfactory results obtained.

"Churning."—For a heavy working boat with bluff lines, the resistance of the water in front is so great that a propeller with a high pitch ratio would tend merely to rotate the water and throw it off the ends of the blades instead of forcing it directly astern. This action is called "churning," and in the case of a gas engine drive is usually due to the attempt to adapt the propeller to the speed of the engine instead of to the lines of the boat.

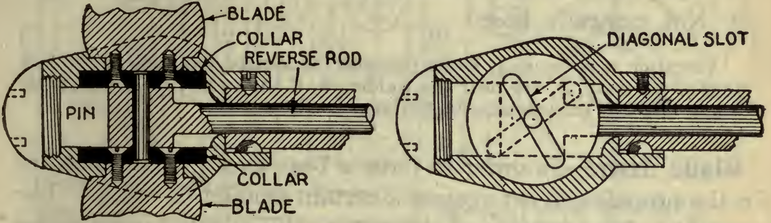


FIG. 3,271*w*.—Reversing propeller. ***In construction,*** the shaft is hollow to receive a reversing rod on the end of which is a pin. The pin works in diagonal slots cut into collars, free to turn in the hub and to which the blades are attached. ***In reversing,*** when the reverse rod is given a fore or aft motion the pin travels in the diagonal slots thus turning the blades and reversing the pitch. ***The adaptation,*** of reversing propellers is for very small boats and can be regarded as not much more than a *makeshift*, as it is difficult to make a reversing propeller strong enough to withstand unusual shocks.

Effect of Dead Wood or Skeg.—Another cause for slip is due to the effect of the skeg in obstructing the free passage of the water to the propeller.

The water is broken considerably before reaching the wheel, and accordingly is not of sufficient solidity for the wheel to act upon efficiently. To overcome this the skeg is usually omitted, especially on motor boats, and a strut installed, thus the wheel is in "solid" or *live* water as it is called.

In many cases the substitution of a strut instead of a skeg has materially increased the speed of the boat. In boats having a thick skeg a three blade propeller should always be used because two blades are always working when the other is in the vertical position.

In skeg boats, the wheel should be as large as possible.

Vibration.—There are only three possible defects in a propeller which would cause vibration:

1. Not properly balanced;
2. Pitch not the same in all blades;
3. Not centrally bored.

Vibration may occur: 1, if the propeller be located too close to the bottom of the boat, 2, too near the rudder, 3, if the rudder be not properly hung, and 4, if the propeller be too large for the engine.

Blade Area.—In order to force a boat ahead, it is necessary for the propeller to act against a certain amount of water. The area of the cross section of the water column acted upon varies in proportion to the diameter of the propeller and the area of the blades.

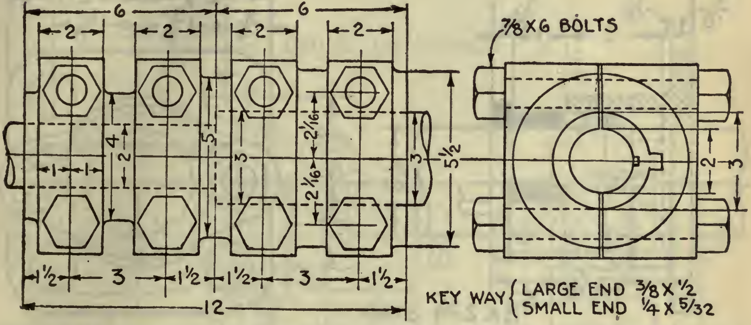
In the case of two propellers of the same diameter, it is readily seen that the one having the larger area acts against a greater quantity of water. A narrow bladed propeller will, therefore, operate at a slightly higher number of revolutions than a wide bladed one of the same diameter and pitch, provided both are of the same general design.

The determination of proper blade area is difficult for the amateur, as there are no hard and fast rules to govern it.

In general, a heavy or hard driving boat should have a propeller with a large blade area, especially if the engine turn less than 600 *r.p.m.* Over 800 *r.p.m.*, a wide blade is likely to churn. In general also the lighter the boat, and the faster the engine turns, the smaller should be the blade area.

A hydroplane is usually a hard driving boat until it starts to plane. Accordingly it is usually necessary to use a wide blade propeller to force it to plane. When it planes it drives easily and the propeller turns at extreme speed. If, therefore, a narrow blade propeller could be substituted after it starts to plane, better speed would undoubtedly result.

Squatting.—When the stern of a boat squats or "suck sdown," it can sometimes be overcome by installing a propeller with narrow blades and low pitch ratio. Squatting, however, is usually due to the stern of the boat being too narrow to offer a sufficient amount of buoyancy when the boat is in motion. Squatting to a marked degree from this cause is noticed in the New England V stern dory.

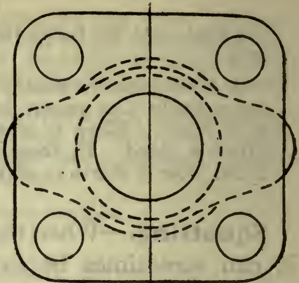


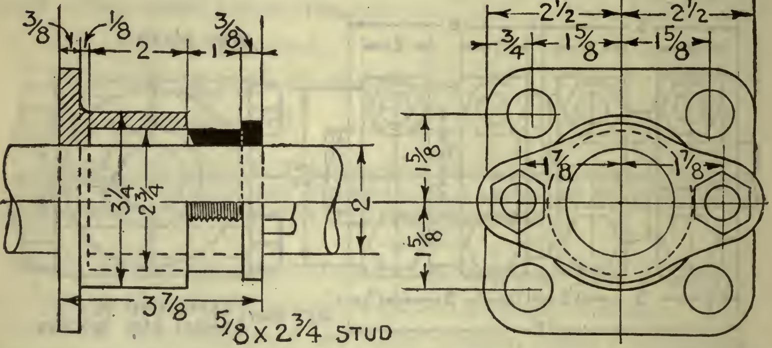
Figs. 3,271x and 3,271y.—Shaft coupling as designed for steamer ***"Stornoway II,"*** consisting of two castings of square and cylindrical section, provided with a key and held firmly together around the shaft by numerous bolts.

The use of "squat boards" has been moderately successful in some cases.

Effective Area of Propeller.—This is the sectional area of the stream of water laid hold of by the propeller, and is generally, if not always, greater than the actual area or area of a circle whose diameter equals diameter of the propeller.

The ratio of effective area ÷ actual area is on good ordinary examples about 1.2 and sometimes as high as 1.4; a fact (according to Rankine's Ship Building, *p* 89), probably due to the stiffness of the water, which "communicates motion laterally amongst its particles."*





Figs. 3,271z to 3,273.—Inside stuffing box as designed for steamer ***"Stornoway II."*** This is virtually the same box as shown below for ***"Stornoway I,"*** except that the gland is adjusted by two bolts instead of a screw follower. ***Bolt adjustment*** *avoids the possibility of flooding in case follower backs off the thread.* The four holes for lag screws should be bored a little larger than the screws to permit adjustment in lining up the shafting.

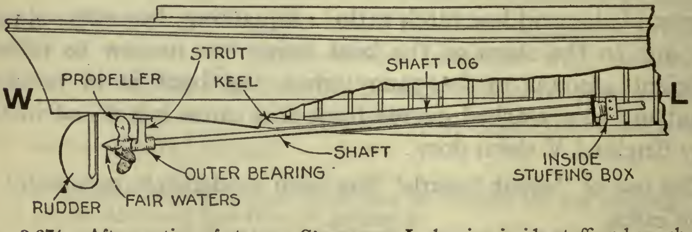


Fig. 3,274.—After section of steamer ***Stornoway I,*** showing inside stuffing box, shaft log, outer bearing, strut, shaft, propeller and fair waters.

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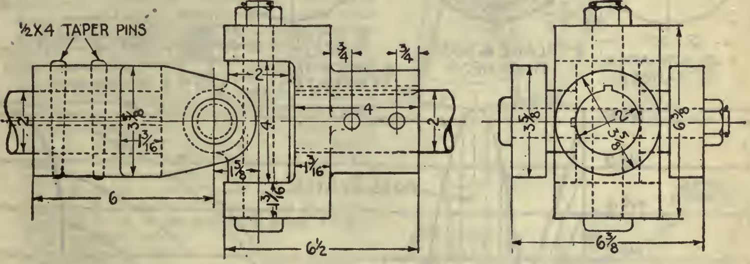
*NOTE.—Prof. Jacobus (*Trans. A. S. M. E.* xi. 1028) found the effective area ratio to vary from 1.2 for steamer Homer Ramsdell with ordinary true pitch propeller, to 1.53 for ferry boat Bergan at speed of 12.09 miles per hour, ordinary true pitch propeller.

Example.—Determine dimensions of propeller for steamer ***Stornoway II,*** with power plant consisting of the engine of Chap. 52, and boiler of Chap. 54, for operation under the following conditions: 42 horse power; 350 *r.p.m.*; speed 12 miles per hour; 15% slip.

A speed of 12 miles per hour, corresponds to

$$\frac{12 \times 5{,}280}{60} = 1{,}056 \text{ ft. per minute}$$



FIGS. 3,275 and 3,276.—Universal joint shaft coupling as designed for steamer ***Stornoway II.*** Since a boat is not a rigid structure, it is continually changing its shape, especially in rough water, hence the engine shaft and propeller shaft cannot be expected to remain in alignment To allow for this and avoid the excessive friction caused by the two shafts being out of alignment a ***universal joint should be used,*** although this refinement is not provided in ordinary practice.

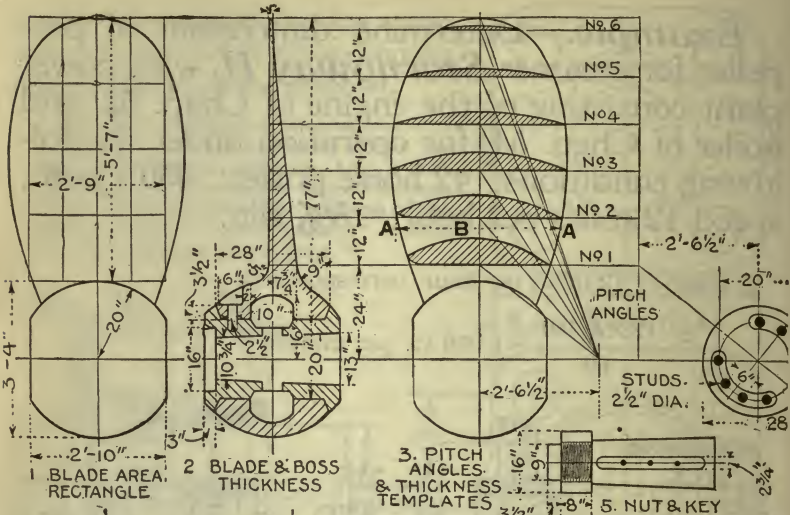
or

$$1{,}056 \div 350 = 3.017 \text{ ft. per revolution}$$

hence with 15% slip

$$\text{pitch} = (3.017 \times 12) \times 1.15 = 41.6, \text{ say 42 ins.}$$

To determine the diameter, the designer must carefully consider the lines of the boat to fix a value for the pitch ratio which

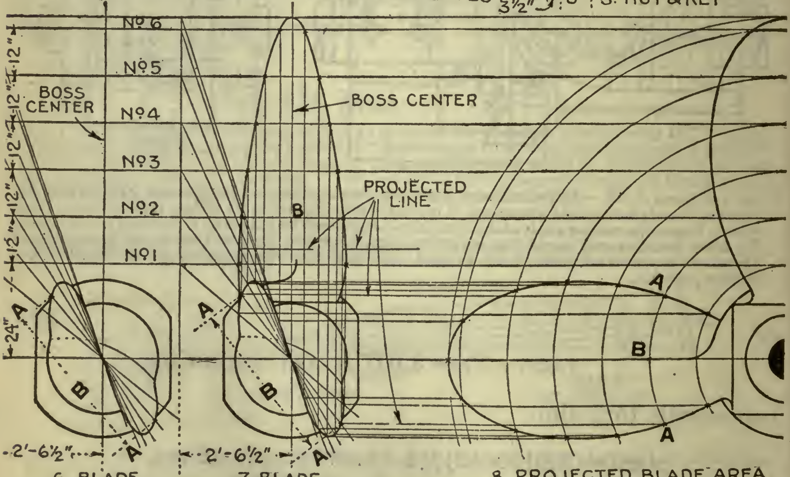


1 BLADE AREA RECTANGLE

2 BLADE & BOSS THICKNESS

3. PITCH ANGLES & THICKNESS TEMPLATES

5. NUT & KEY



6. BLADE PROJECTION

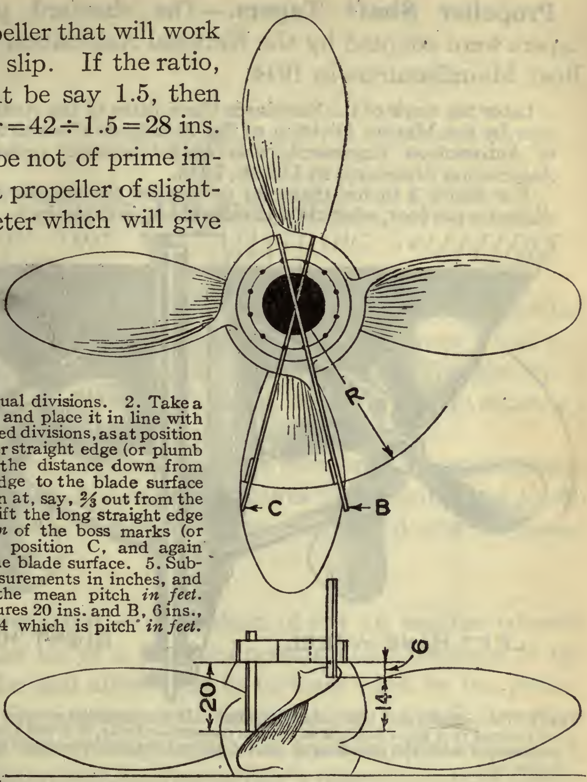
7. BLADE PROJECTION

8. PROJECTED BLADE AREA (ACTUAL THRUST AREA)

Figs. 3,276a to 3,276h.—Drawings of 14½ ft. × 16 ft. built up propeller. Expanded blade area 63 sq. ft.; pitch ratio 16 ÷ 14½ = 1.1; area ratio, 63 ÷ $(14\frac{1}{2})^2 \times .7854 = .38$.

will give a propeller that will work with the given slip. If the ratio, in his judgment be say 1.5, then diam. propeller = 42 ÷ 1.5 = 28 ins.

If draught, be not of prime importance, use a propeller of slightly larger diameter which will give less slip.



FIGS. 3,276*j* and 3,276*k*.—Pitch with propeller horizontal on the floor. ***1st method:*** 1. Mark off the face of the boss into twelve equal divisions. 2. Take a long straight edge, and place it in line with two oppositely placed divisions, as at position B. 3. Take another straight edge (or plumb line) and measure the distance down from the long straight edge to the blade surface at a radius R, taken at, say, ⅔ out from the boss center. 4. Shift the long straight edge through *one division* of the boss marks (or $^1/_{12}$), as shown at position C, and again measure down to the blade surface. 5. Subtract the two measurements in inches, and the difference is the mean pitch *in feet*. That is, if C measures 20 ins. and B, 6 ins., then 20 − 6 = 14 which is pitch *in feet*. This method can be applied to the propeller in its usual position on the shaft

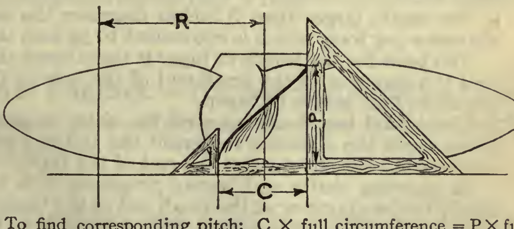


FIG. 3,276*i*.—Pitch with propeller horizontal on the floor ***2nd method.*** Fit up two set squares and a horizontal piece of wood as shown, at a suitable radius R (say about two-thirds out from boss to give average pitch); then the piece of pitch P, will be obtained by the vertical measurement, and the piece of circumference C, by the horizontal or floor measurement. And R × 2 × 3.1416 = full circumference at radius R. To find corresponding pitch: C × full circumference = P × full pitch, from which full pitch = full circum. × P ÷ C. If the blades have a *varying* pitch, repeat the above at two or three radial positions, and take the *average* pitch.

Propeller Shaft Tapers.—The standard propeller shaft tapers were adopted by the National Association of Engine and Boat Manufacturers in 1914.

Later the work of the Standards Committee of the Association was taken over by the Marine Division of the Standards Committee of the Society of Automotive Engineers, who quite radically revised the National Association Standards in March, 1918.

For shafts 3 inches diameter or smaller the standard taper is ¾ inch diameter per foot, which is equivalent to $\frac{1}{16}$ inch per inch.

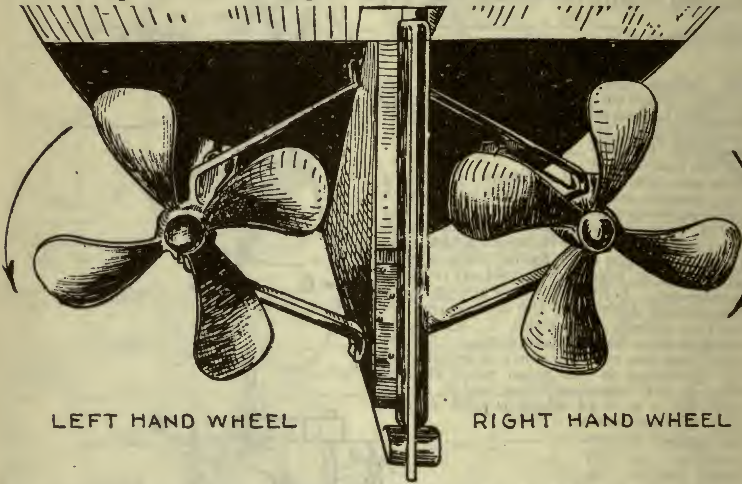


Fig. 3,276*l*.—Right and left hand propellers. If the propeller revolve *clockwise* to force the boat ahead it is right hand; if it revolve *counter-clockwise*, it is left hand. Propellers should be installed with the rounding or convex side of the blade toward the engine, the flat side facing aft.

For shafts larger than 3 inches diameter the standard taper is 1 inch diameter per foot, which is equivalent to $\frac{1}{16}$ inch in ¾ of an inch.

The basis for the length of taper is three times the diameter of the shaft, and the diameter of the small end of the hole in the propeller hub corresponds to that length of taper.

The actual length of taper on the shaft is made slightly shorter than three times the diameter to permit the tail nut to be screwed tight without striking the shoulder at the end of the taper.

In tapering shafts and boring propellers it is necessary to work accurately to the diameter at the small end of the taper, and it should be noted that the diameter of the small end of the hole in the hub of the propeller should be smaller than the diameter of the shaft at the small end of the taper.

CHAPTER 53

HOT AIR ENGINES

The Medium.—The utilization of the expansion of heated air for driving an engine has for many years been a subject which has exercised the ingenuity of inventors. The history of hot air engines includes many failures; there are certain constructive difficulties which have been hard to overcome.

As a medium for transforming heat into work, air can be used with safety at much higher temperatures than steam, and therefore a larger proportion of the heat given to it can be transformed into work.

Cycle of Operation.—The action of hot air engines consists in admitting the air at a high temperature and pressure to the working cylinder and allowing it to perform work on the piston, thus reducing its pressure and temperature, when it is either exhausted into the atmosphere and a fresh supply introduced, or it is again compressed, and heated for a repetition of the former process.

Classes of Hot Air Engines.—The principal varieties of hot air engine may be classified according to the nature of the working cycle, as those in which:

1. Changes of temperature take place at a pair of constant volumes;

2. Changes of temperature take place at a pair of constant pressures;

3. Heat is received and rejected at a pair of constant pressures.

The first two classes, fitted with *economizers* are in theory *perfect engines*, that is, they are theoretically capable of transforming into work the largest fraction of the heat received from the fuel that the limits of temperature allow. Those of the third

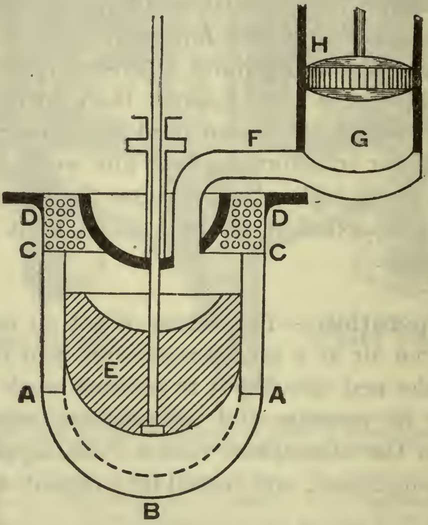


FIG. 3,276*m*.—Diagram of hot air engine (the type improved on by Rev. James Stirling). ***In construction,*** DCABACD is the air receiver or heating and cooling vessel; G, cylinder with its piston H. The receiver and cylinder communicate freely through nozzle F, which is always open during operation. Within the receiver is an inner receiver or lining of similar figure, as B to CC. The hemispherical bottom of this lining is pierced with many small holes, and the space between it and the bottom of the outer receiver is vacant. From AA to CC, the annular space between the outer receiver and its lining contains the regenerator, being a grating composed of a series of thin vertical oblong strips of metal or glass, with narrow passages between them. The inner surface of the cylindrical part of the lining, from AA up to CC, is turned, and the plunger E, moves vertically up and down within it, fitting easily, so as to leave the least space possible without causing perceptible friction. This plunger is hollow, and filled with brick dust, or some such slow conductor of heat. The space from CC to DD, between the barrel of the receiver and the concave part of its cover, and class are not perfect engines, but possess some practical features which will afterwards be explained.

Most hot air engines are very bulky for the power developed because they are so designed that the original pressure of the air which is compressed and heated is that of the atmosphere. If, however, the working air be confined in the machine, and originally compressed to a high pressure, this difficulty disappears.

Thus, assuming it practicable to maintain a temperature in a given air engine sufficient to double the original pressure of the air, then, if the air be admitted at the pressure of the atmosphere, the available pressure, after heating, would be 14.7 lbs. per square inch, gauge pressure.

Again, if the supply of air be drawn from a reservoir, in which the pressure be 60 lbs. per square inch, the effect of increasing the temperature to the same point as in the former case would be to double the original pressure, making it 120 lbs. per square inch. It seems strange that the majority of inventors should have ignored this significant principle, and that, too, in the face of the example afforded by the Sterling engine (one of the first air engines), which from all accounts seems to have been very successful and economical.

Class One.—To illustrate the first division of the classification, the engine above mentioned is given as an example. It was invented by the Rev. Dr. Sterling in 1816, and subsequently improved by him and his brother James Sterling of Edinburgh. The same mass of air was used again and again, and compressed at starting to a pressure of 7 to 10 atmospheres.

FIG. 3,276*m*.—*Text continued.*

above the upper edge of the lining, contains the refrigerator, which consists of a horizontal coil of fine copper tube, through which a current of cold water is forced by a pump, not shown in the figure. There is an air compressor, not shown, which forces into the nozzle F, enough of air to supply the loss by leakage. The hemispherical bottom ABA, of the receiver forms the heating surface which is exposed to the furnace. The effect of the alternate motion of the plunger E, up and down is to transfer a certain mass of air, which may be called the working air, alternately to the upper and lower end of the receiver, by making it pass up and down through the regenerator between AA and CC. The perforated hemispherical lining of the bottom of the receiver causes a diffusion and rapid circulation of the air as it passes into the lower end of the receiver, and thus facilitates the convection of heat to it, for the purpose of enabling it to undergo expansion, during which expansion it lifts the piston H. The descent of the plunger causes the air to return through the regenerator to the upper end of the receiver It leaves the greater part of the heat corresponding to the range of temperature T^1—T^2 stored in the plates of the regenerator. The remainder of that heat (being the heat wasted by the imperfect action of the generator) is abstracted by the refrigerator, which also abstracts the heat produced by the compression of the air when the piston H, descends. The heat stored in the generator serves to raise the temperature of the air, when, by the lifting of the plunger E, it is sent back to the lower end of the receiver. The mechanism for moving the plunger E, is so adjusted that the up stroke of that plunger takes place when the piston H, is at or near the beginning of its forward stroke, and the down stroke of the plunger when the piston H, is at or near the beginning of its back stroke.

In construction, a cylindrical air receiver, in which a plunger could be moved up and down, was placed over the fuel of the furnace. The annular space between the plunger and the sides of the receiver was occupied by an immense number of thin sheets of metal, which formed the economizer. In the upper part of the receiver, which communicated with one end of a working clyinder of the usual construction was a *refrigerator*, consisting of a coil of tubing through which cold water circulated. The plunger was alternately raised and lowered by suitable mechanism, and in its motion caused the great body of air in the machine to occupy alternately the heating end, and the cooling end. Power was thus produced by the alternate expansion and contraction of the air.

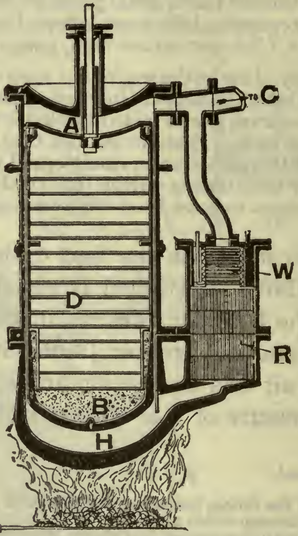


FIG. 3,276*n*.—Stirling's hot air engine showing improved heater and displacer. This engine worked with high pressure up to 150 lbs., the leakage being made up constantly by a small pump. ***In construction,*** C, is the pipe, to working cylinder, A, the cold end, and H, the hot end of the displacer cylinder. The regenerator R, consisted of bundles of thin sheet iron, and a refrigerator W, consisted of copper pipes with circulating cold water.

Owing to the great pressure under which the engine was operated, its size for a given horse power was very moderate. It was ultimately abandoned, however, because of the failure of the receiver to withstand the destructive action of the heat.

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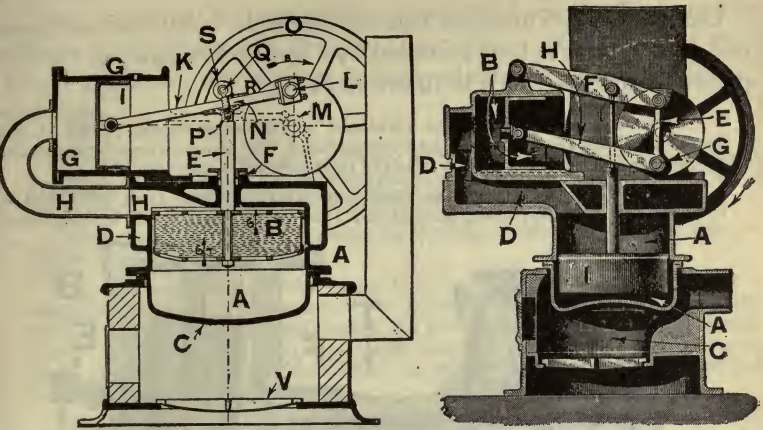
*NOTE.—Ericsson's engine is interesting chiefly on account of the enormous scale on which its construction was actually carried out. The engines of the steamship "Ericsson" had cylinders each 14 feet in diameter and the other parts in proportion. The trials of this vessel were conducted in a manner which did not allow any confidence to be placed in the results said to have been obtained, and steam engines replaced those of Ericsson within two years.



FIGS. 3,276*o* and 3,276*p*.—Sectional views showing working and actual construction of Robinson's hot air engine, a successful English type. In fig. 3,276*o*, A, is the air chamber in which the displacer or regenerator B, is worked. C, is the heater forming the lower part of this chamber. D, is a jacket or water space through which water or other fluid circulates to keep the upper part of A, cool, or a coil of pipe may be used as described hereafter. A rod E, or its equivalent is connected to the displacer or regenerator, and this rod works air tight through the gland F, or its equivalent. G, is the working cylinder in communication with the chamber A, by the pipe or passage H, so that the variations of pressure in A, act upon the piston I, working in the cylinder. The piston I, is connected by a connecting rod K, to the crank or disc pin L, and shaft M. The upward or downward movement of the crank pin L, is communicated to the displacer or regenerator B, by means of the connecting rod K, the head N, and the rod E, while at the proper times the horizontal movement of the piston I, drives the crank pin L, shaft M, and fly wheel O, The connecting rod K, works through the head N; this head N, is jointed into the rod E, by the pin P, and a pin Q, with adjustable bolts R, or their equivalent connect a friction pulley or its equivalent S, into the head N, to run or roll upon the connecting rod K, to reduce friction. A coil of pipe may be used at D, to cool the upper part of the chamber A. V, is the fire grate shown with fire bars, but a burner may be used for liquid or gaseous fuels. The displacer or regenerator B, is formed as a metal casing, preferably of iron or steel, with openings in it for the passage of air in the directions as shown by arrows 6, and this casing is filled with wire gauze, or other suitably perforated material, through which the air will pass and lose its heat on the down stroke and take up heat on the up stroke. The actual construction of the connecting rods are shown in fig. 3,276*p*.

NOTE.—The following results of hot air engine tests are given by Bryan Donken:

Engine	Cylinder	R. P. M.	I. H. P.	B. H. P.	Lbs. of fuel	
					I. H. P.	B. H. P.
Buckett	24×16	61	20.2	14.	1.8	2.5
Beniér	13.4×13.8	117	5.8	4.	3.1	3.6
Bailey	14.6× 6.9	106	2.4	1.3	4.2	7.6
Rider	6.7×95	138	.81	.23		

Class Two.—The most familiar example of the second class of hot air engine is that invented by Captain Ericsson, and which differed from Sterling's in many respects.

In this engine fresh air was admitted to the compression pump at the lowest working temperature, and compressed, the temperature being maintained constant by the action of some refrigerating apparatus.

The air when compressed entered a receiver. It was then admitted to the working cylinder, being heated in its passage to the higher temperature, so that its volume was increased, and the pressure kept constant under the movement of the piston.

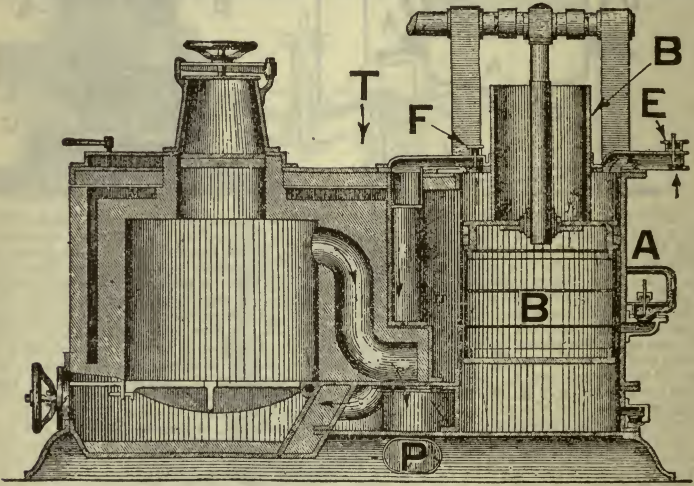


FIG. 3,276*q*. Shaw hot air engine. There are two working cylinders A, each single acting, and the compressing pumps are formed by trunks on the upper sides of the pistons. B is one of the pistons and B′ the corresponding trunk. The regenerator is in the rear not visible. Air is drawn in by the compressing pump through the valve E, and forced into the regenerator and furnace through the valve F. P is the exhaust pipe.

During expansion, the temperature of the air was maintained constant at the higher limit, after which the air was expelled into the atmosphere giving up its heat to the economizer to be used in heating the volume next introduced.

Class Three.—An example of the third class of hot air engines is that invented by Philander Shaw. Its chief feature is the use of the products of combustion themselves, instead of merely the air heated by them to drive the piston. Much heat is thus saved which in other engines is unavoidably sent up the chimney and lost.

In construction, the working piston of the Shaw engine is fitted with a trunk on its upper side, which, thus reduced in area, serves as a compressing pump, and the products of combustion act directly upon its other side.

The enclosed furnace has means for feeding it without allowing any air to enter.

The air compressed by the pump is delivered into the furnace where it combines with the fuel to form the gases of combustion, thus receiving additional heat, expands and raises the piston for a portion of the stroke. The admission valve is then closed, and the gases expand without additional heat, until the piston has completed its stroke, and are then discharged into the atmosphere.

The addition of air economizer greatly increases the efficiency of this type of engine but its principal advantage is that by using the products of combustion inside the engine, much heat is saved which in other engines is unavoidably sent up the chimney and lost.

The Economizer and Refrigerator.—If in the operation of a hot air engine, all the heat which is necessary to raise the air to the required temperature be thrown away or lost every time the air is cooled, the efficiency of the engine is greatly reduced. An *economizer* therefore is provided, the object of which is to store up the heat rejected by the air when it falls in temperature and subsequently to raise the temperature of the air by restoring the same heat, so that the only heat which the furnace has to supply is the latent heat of expansion, together with the amount of sensible heat which may be lost through the imperfection of the economizer.

When in the operation of the engine it is necessary to cool the air after it has been brought to its greatest heat it is not at once brought into contact with the coldest part of the vessels. This would effectually cool it, but the heat when thus extracted would be entirely lost, because it could never again be taken up by a body warmer than itself. To avoid this loss, the cooling of the air is accomplished in two stages by means of:

1. The economizer; and,
2. The refrigerator.

The highly heated air passes from the cylinder to the economizer which consists of a multitude of narrow passages, whose temperature at the cylinder end is nearly as great as that of the hot air, but which gradually declines till at the end adjacent to the refrigerator it becomes nearly as low as the refrigerator. The latter is simply a cylinder in which the compression plunger works, and which is surrounded by a jacket through which cold water circulates to cool the air to the lowest temperature of the cycle.

When two bodies of unequal temperature come in contact or near each other, a transfer of heat will take place from the hot body to the colder body, the air, therefore, when it enters the narrow passages of the economizer, will give out a portion of its heat even to the hottest part of these passages, and in its progress through the economizer will continue to give out more and more heat as the temperature of the passages is diminished, till finally when it is ready to escape into the refrigerator, there remains only a small portion of the heat to be extracted. This is absorbed by the refrigerator. The greater part of the heat, therefore, has been left behind in the thin sheets of metal which form the passages of the economizer, and which are so arranged as to retain the heat until the now cold air in the refrigerator is compressed and sent back through the economizer to the cylinder.

The cold air on its return through the economizer comes into contact with metal that is hotter than itself and absorbs progressively, most of the heat which it gave out to the economizer during its previous passage. In passing from the economizer to the cylinder, therefore, the air requires but a small amount of heat to give it the elasticity necessary for the production of power. Hence, instead of requiring for each power stroke as much heat as would be sufficient to raise the air from its lowest to its highest temperature, it is necessary to furnish only as much as will heat it in the same number of degrees by which the hottest part of the air vessel exceeds the hottest part of the intermediate passages.

Use of Hot Air Engines.—The practical application of this type of engine is where the power demands are light, and where there is an abundance of cooling water for the refrigerator. The hot air engine is confined to small powers on account of its great bulk in proportion to the power developed. The particular field of its usefulness is in light service pumping installations, as in suburban homes, or apartment houses where the pressure of the city water supply is insufficient for the upper floors. All the water pumped passes around the refrigerator, hence the air is cooled without expense.

The small fuel consumption, and automatic operation are desirable features; where gas or oil fuel is used, no attention is required after starting. A few examples of modern hot air engines will now be given.

The Reeco-Ericsson Engine.—A sectional view of this engine is shown in fig. 3,276*r*. There is a single cylinder 1, in which is a power piston 2, and a transfer plunger 3.

NOTE.—In a test of vertical double cylinder hot air engine, the results obtained gave 20.19 I.H.P. in working cylinder and 11.38 I.H.P. in the pump, leaving 8.81 net I.H.P. The effective brake H.P. was 5.9, giving a mechanical efficiency of 67 per cent. Consumption of coke, 3.7 lbs. per brake H.P. per hour. Mean pressure on pistons 15.37 lbs. per sq. in., and in pumps 15.9 lbs., the area of working cylinders being twice that of the pumps. The hot air supplied was about 1,160° Fahr., and that rejected at the end of the stroke about 890°.

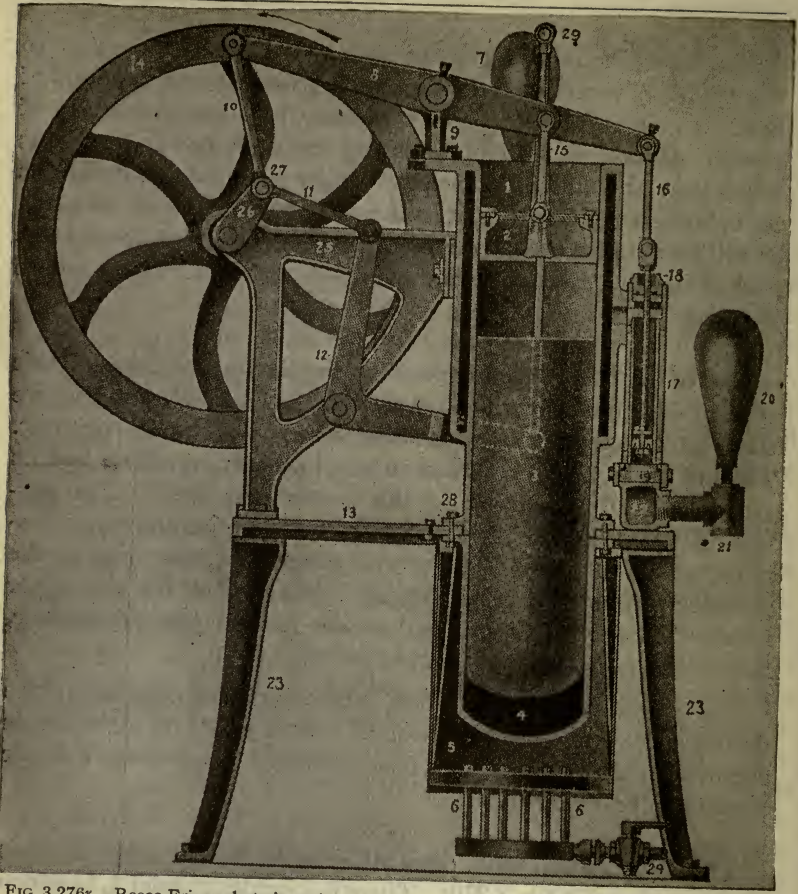


FIG. 3,276*r*.—Reeco Ericson hot air engine, 1, cylinder; 2, air piston; 3, transfer piston; 4, heater; 5, furnace; 6, gas burners; 7, air chamber; 8, main beam; 9, beam center bearing; 10, connecting rod; 11, bell crank link; 12, bell crank; 13, bed plate; 14, fly wheel; 15, air piston links; 16, pump link; 17, pump chamber; 18, pump gland; 19, suction valve; 20, vacuum chamber; 21, suction pipe; 22, pump bottom; 23, legs; 24, gas cock; 25, crank shaft bracket; 26, crank; 27, crank pin; 28, heater bolts; 29, transfer piston rod cross head. ***In operation,*** after the lower end of the cylinder has been sufficiently heated, which takes only a few minutes, the engine is started by hand by giving the wheel one or two revolutions. The air contained in the machine is first compressed in the cold part of the cylinder; it is then transferred to the lower end, where it is heated and expanded, thus furnishing the power. This engine, like all other hot air engines, is single-acting, the momentum of the fly-wheel continues the revolution until the next impulse; the cycle is completed in one revolution. The same air is used continuously.

The piston receives and transmits the power, and the plunger transfers the air contained in the machine alternately, and at the proper time, from one end of the cylinder to the other.

The cylinder is surrounded at its upper end with a water jacket, or refrigerator, through which all the water passes on its way from the well to the tank. This keeps the upper end of the cylinder cool, while the lower end is exposed to the fire and becomes as hot as it is practicable to make it.

By the *peculiar arrangement* of connections between the air and transfer pistons the proper relative motions between these pistons are obtained.

The engine operates as follows: After the lower end of the cylinder has been sufficiently heated, the engine must be started by hand by giving it one or two revolutions. The air contained in the machine is first compressed in the upper end or cold part of the cylinder; it is then transferred to the lower end where it is quickly heated and expanded, thus furnishing the power.

This engine, like all other hot air engines, is only single acting. The momentum of the fly wheel continues the revolution until it receives an additional impulse by the repetition of the above mentioned conditions, which occur once in every revolution. The same air is used continuously, and is cooled, compressed, heated and expanded in the regular order and without noise.

Performance of Reeco-Ericsson Engines.

Diameter of cylinder.	Cubic ft. of gas per hour.	Quarts of kerosene per hour.	Pounds of anthracite coal per hour.	Gallons per hour for total head of 50 feet.
5 inch	12	1	4	150
6 "	16	2	4	300
8 "	20	2	5	500
10 "	50	3	6	1.000

Directions for Setting and Running the "Reeco" Ericsson Engine.—Place the engine as near the water as possible. The engine, except when a deep well pump is used, should be so placed that the vertical distance from the foundation on which it stands to the surface of the water is not over twenty feet. These engines will draw water as high as any fast running pump, but it is always good practice to set a pump within twenty feet (vertical) of the water where it is possible to do so.

Foundation.—The engine should stand level, and any kind of a foundation may be used. In cases where coal or wood furnaces are to be used, the foundation should be made of either brick or cement.

Water Pipes.—The pipes should be ample in size. The size that these engines are tapped for is as follows: ¾ inch for 5 inch engine; 1 inch for 6 inch engine; 1¼ inch for 8 inch engine, and 1½ inch for 10 inch engine. These sizes will answer, provided the pipe is not too long; but when the pipe is long, and perhaps crooked, these sizes should be increased one size. In piping up these engines, always place unions close to the engine, both on the suction and discharge pipe, and also, when using gas, on the gas pipe. Also arrange for draining these pipes, so that all the water may be drawn off to prevent damage by freezing.

Gas Pipe and Operation of Burner.—In setting up engine with gas furnace, it is advisable to have gas pipe sufficiently large, in order to keep the pressure of the gas up to its proper point when it is delivered to the burners, otherwise the burners will not work satisfactorily. The proper size pipe is ½ inch for 5 inch and 6 inch engines; ¾ inch for 8 inch and 10 inch engines. These sizes will answer unless the pipes be long, when they should be increased.

Connect the burner with the gas pipe and put a cock in the gas pipe within one foot of engine. Regulate the supply of gas by opening this cock until the proper flame is obtained. There are two holes in the neck of the burner and two corresponding holes in the movable slide on the brass nipple. When lighting the gas or turning it off, the proper position of this movable slide is with the holes in the gas burner neck covered by the solid part of the slide. After the gas is lighted, this slide should be turned so that the holes in the gas burner neck are opposite the holes in the gas burner slide and sufficiently opened to admit the proper quantity of air to have the gas burn with a blue flame.

The right heat is obtained from a blue flame and the engine should start in from ten to thirty minutes after lighting the gas. A little observation of the flame will teach the attendant how to regulate the flame so as to start the engine in the shortest time.

Stove Pipe and Draft.—A good draft is necessary, particularly when burning coal. The proper size pipe is 5 inch for 5, 6 and 8 inch engines, and 6 inch for 10 inch engine.

For burning wood it is not necessary to have the draft so sharp as for burning coal. The best coal to use is anthracite, chestnut size. If the engine is to be placed away from the house and a small house built over it, the stove pipe should run up about 10 feet, more or less, depending upon the nature of the surrounding objects, such as trees and other houses.

The object is to secure a good draft at all times and to have nothing interfere with it, no matter in what direction the wind may blow. A damper must be put in every stove pipe.

Firing.—*Coal.*: The fire should be thin and bright; too much coal should not be put on at once, but rather a small quantity and oftener. This will keep the fire at a uniform heat and the engine at a uniform speed. Do not overheat the bottom of the engine. This may be determined by looking through the fire door and shading the light of the fire from the bottom of the engine by a fire shovel or something of that kind. This part of the engine should not be heated above a dull red heat. Wood.: Any kind of wood may be used, and it is difficult to damage the machine by overheating with wood fuel.

Speed.—The best speed to run these engines at is 100 to 120 revolutions per minute for the 5 inch and 6 inch engines, and about 80 to 110 revolutions per minute for the 8 inch and 10 inch engines.

Oiling.—All the working parts of these machines need a few drops of oil every time the engine is run. Do not use much oil on the cylinder. A swab is furnished with every engine and should be used for oiling the inside of the cylinder instead of applying the oil from the squirt can. A few drops of oil should also be put around the transfer piston rod where it works through the leather packing.

Oil.—Special oil for use on these engines can be obtained from the manufacturer, however, a mixture of one part sperm oil with two parts of good paraffin oil will answer if used sparingly.

Starting.—Start the engine as soon as it is hot enough to run. Be particular about this, otherwise the engine will be unduly heated, and perhaps damage done. While the engine is running, the heat developed by the fire is being utilized to heat the air in the machine; but while the engine is standing this is not going on, and consequently the danger of over heating is much greater than when the engine is running. To start the engine, turn the wheel by hand in such a direction that the top of the wheel will turn from the top of the cylinder.

If the engine be hot enough to run it will only be necessary to turn it by hand about two revolutions, when it will go without any more assistance.

The blow off cock should be left open while the engine is being heated up, and the engine turned one or two revolutions before closing this cock, in order to expel any moisture, etc., that may have collected inside of the cylinder while the engine has been standing idle. Shut the cock when the main piston is at the top of its stroke.

Stopping.—To stop the engine, open the blow off cock on the side of the cylinder and, unless the engine is very hot, it will stop after making three or four revolutions. This cock should be left open until it is desired to start the engine again. The engine should not be stopped for any length of time when the fire is burning brightly under it. If it become necessary to keep the engine standing, cover the fire over with a little fresh coal and leave the fire door partly open, so as to protect the bottom of the engine from becoming unduly heated. The engine should be turned, after it has stopped, to such a position that the transfer piston will be at the top of its stroke, and left standing so until it is started again.

Packing.—The packings of these engines are leather and consist of one large packing in the main or air piston and one small cup shaped packing on the transfer piston rod. To put on a new air piston packing it should be first well soaked in oil and then thoroughly worked in the hands until it becomes soft and pliable. Remove the air piston from the engine, take off the old packing and replace it with the new one with the flesh side of the leather upward and screw all the bolts down securely. Place the air piston at the top of the cylinder and work the edge of the leather down into the cylinder being careful to avoid any puckers or unevenness which would allow the air to escape. This operation requires only a little care to do it successfully.

The packing on the transfer piston rod is pressed in a mould to the proper shape. Do not oil this packing before putting it on. Screw it over the threads on the end of the transfer piston rod with the raised side upward. After passing the threads, unscrew the packing, remove it, reverse it and screw it down to its proper position, then oil it. This method insures the packing fitting tighter and lasting longer.

The pump rod packing is also moulded to the proper shape and has a raised center. The same method of putting on the pump rod packing should be followed and no oil is needed.

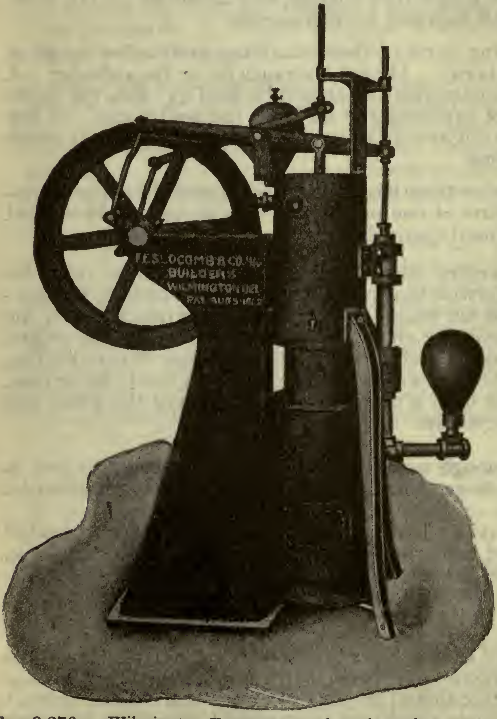
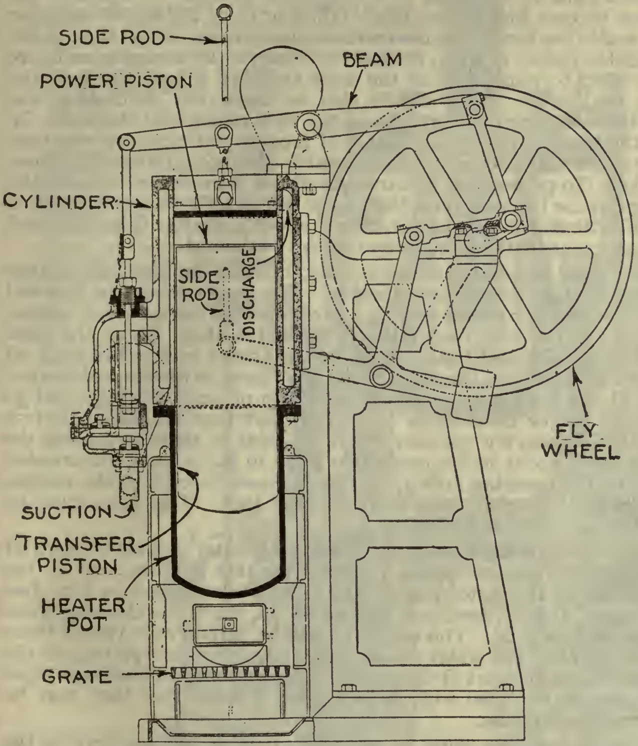


Fig. 3,276*s*.—Wilmington Ericsson type hot air engine.

Cleaning.—When the



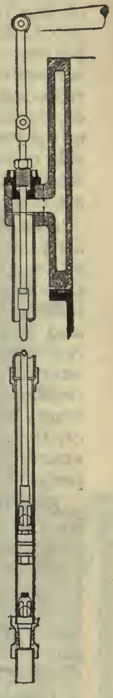


FIG. 3,276*t*.—Sectional view of Denney-Ericsson hot air engine with names of parts.

FIG. 3,276*u*.—Denney-Ericsson hot air engine deep well pump. When the water is lower than say 20 ft. below the surface of the ground it is necessary to use a deep well pump as here shown. In this case the engine must stand close to the well and in such a position that the pump rod will go directly from the engine down to the pump. When it is possible to do so, these pumps should be lowered *into the water*, so as to avoid any work on the suction side. The pipe which connects the pump with the engine is made sufficiently large for the pump rod to work inside of it, and also to allow sufficient space for the water to pass upward. It is generally preferable to cut this discharge pipe into short lengths—say, from eight to ten feet—as they are much more easily handled. The pump rod is made of pipe or wooden rods, as may be deemed best. These sections of pipe and rod are screwed together as they are lowered down, it being unnecessary to go into the well. ***In general practice*** the extreme depth to which pumps may be lowered is: 6 in. engine, 30 ft.; 8 in. engine, 60 ft.; 10 in. engine, 135 ft. It is not usual to lower pump on the 5 in. engine.

engine shows signs of weakness it is frequently because the inside of the cylinder has become foul by too much oiling or from the use of improper oil. As a scale thus formed on the inside of the cylinder and on the outside of the transfer piston is hard, it must be removed by scraping and care should be taken to remove all of the scale and dirt which falls into the bottom of the heater. Care should also be taken not to damage the surface of the cylinder or the transfer piston by scraping into the iron. In replacing the transfer piston, see that it is properly adjusted by the nuts on the top of the transfer piston rod so that it does not strike on the down or up stroke. To properly adjust transfer piston, turn the wheel until the side rods are at the highest point, then draw the transfer piston up until it touches the bottom of the air piston, drop it about a quarter of an inch and set it at that point, with the nuts on the top of the rod.

Heater.—If the heater should be cracked, it becomes apparent through the air blowing through it down into the fire. The heater should be renewed at once, or the transfer piston may be destroyed. To put on a new heater, first remove the furnace; then take off the old heater. Make the joint between the cylinder and heater tight, by using red lead mixed with oil and spread it evenly and thin all around the recess in the top of heater. Screw up all the bolts evenly by tightening up on opposite bolts and go over all bolts two or three times to be sure that they are tight. Before replacing the furnace, try whether or not the joint be tight by turning the wheel backwards about half a revolution so as to put a little compression on the air in the machine and while this pressure is kept in the engine, examine the joint carefully and listen for any leak. This joint must be perfectly tight.

Freezing.—To prevent damage by freezing, all water should be drawn off in cold weather. Two cocks are provided for this purpose, one on the pump and one on the water jacket, and these cocks should be kept free from sediment, and in condition to use at all times by cleaning them occasionally with a wire. The engine should be turned over two or three times by hand, after the water ceases to run out, to insure getting all the water out of the upper part of the pump. The suction and discharge pipes should always be provided with drain cocks, in order that they may be drained in exposed places.

Leaks.—These engines must be air tight, or they will not develop the maximum power. They are frequently run, however, with a leak in the air piston packing, but it is a bad practice to run them so, and when a leak develops, new packing should be put on. The transfer piston should also be air tight; that is, the piston itself. If for any reason, such as overheating, or damage from any cause, it leak air, the engine will not run, and a new piston will have to be put in before the engine will develop its power.

The Reeco-Rider Engine.—In this engine, compression and expansion take place in separate cylinders. Fig. 3,276*v* is a sectional view showing the working parts.

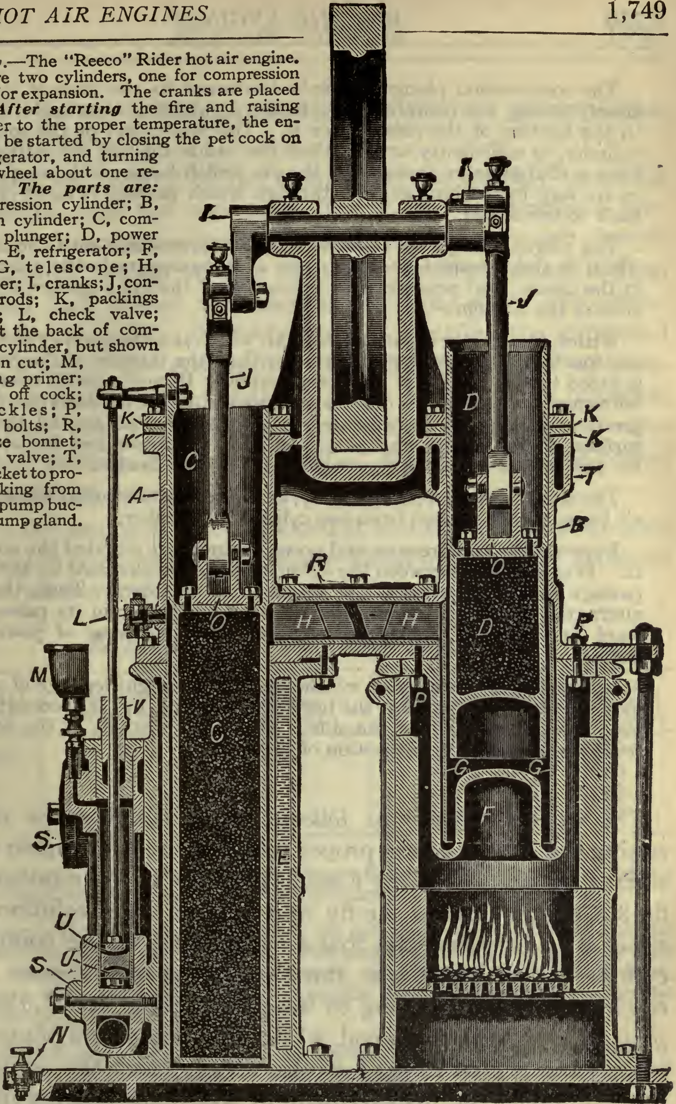


FIG. 3,276*v*.—The "Reeco" Rider hot air engine. There are two cylinders, one for compression and one for expansion. The cranks are placed at 95°. ***After starting*** the fire and raising the heater to the proper temperature, the engine may be started by closing the pet cock on the refrigerator, and turning the fly wheel about one revolution. ***The parts are:*** A, compression cylinder; B, expansion cylinder; C, compression plunger; D, power plunger; E, refrigerator; F, heater; G, telescope; H, economizer; I, cranks; J, connecting rods; K, packings (leather); L, check valve; placed at the back of compression cylinder, but shown at side on cut; M, pumping primer; N, blow off cock; O, knuckles; P, heater bolts; R, economize bonnet; S, pump valve; T, water jacket to protect packing from heat; U, pump buckets V, pump gland.

The compression plunger extends downward to the base of the engine, closely fitting the compression cylinder A, which also extends downward to the bottom of the refrigerator E. The lower part of the compression cylinder, is sufficiently smaller than the inside shell of the refrigerator to form a thin annular passage for the air, which becomes thoroughly cooled on its way to the bottom, and through which passage it flows on its way back to the heater.

The power plunger D, likewise extends downwards into the heater F, which in shape resembles the bottom of a champagne bottle, that is, rising in the center, and presenting to the action of the fire a narrow annulus all around the bottom.

Within this heater is the telescope G, which is a thin iron cylinder, about one-fourth of an inch less in diameter than the interior of the heater. It is fitted to the interior of the power cylinder B, and extends nearly to the bottom of the heater. Its office is to cause the air, which flows from the compression cylinder to be presented in a thin sheet all around the interior surface of the heater, and particularly at the lower and hotter portion. By this means the air is thoroughly and rapidly heated.

The same air is used continuously, as there is neither influx nor escape, the air being merely shifted from one cylinder to another.

Between the compression and power cylinders is situated the economizer H. It is so placed between the cylinders as to be traversed by the air in its passage each way between the *hot* and *cold* cylinders. Thus, the heat is alternately abstracted from, and returned to the air in its passage backward and forward, imparting economy and steadiness of power to the engine.

L, is a simple check valve which supplies any slight leakage of air which may occur. It is placed at the back of the engine, but is necessarily shown (in the sectional cut) on the side. The other portions of the engine are readily understood by inspection of the cut.

The engine operates as follows: After starting the fire and raising the heater to the proper temperature, the engine may be started by simply closing a small pet cock in the economizer H, fig. 3,276*v* and turning the fly wheel about one revolution. The effect is to compress the cold air contained in the compression cylinder C, to about one third of its normal volume. After reaching this point, owing to the cranks being at 95°, the power plunger D, begins to ascend, while the compression plunger C, is completing the downward stroke which forces the air into the heater. Since the displacement in the compression cylinder is equalized by the receding power piston there is no noticeable change of volume.

The sudden heating of the compressed air quickly raises its temperature, and the expansion due to the temperature produces a great increase of pressure, which forces the power plunger to the end of the up stroke. The compression piston is then 5° behind half stroke and moving upward, when the power plunger, by beginning its down stroke, transfers the expanded hot air through the economizer into the refrigerator or compression cylinder, at or slightly below atmospheric pressure; in the latter case the deficiency is supplied by the check valve L. After passing the upper center, the compression piston again begins its descent as before.

When the engine is to be stopped it is only necessary to open the small pet cock on the economizer which prevents the accumulation of pressure.

Performance of Reeco-Rider Engines.

Diameter of cylinder.	Cubic ft. of gas per hour.	Quarts of kerosene per hour.	Pounds of Anthracite coal per hour.	Gallons per hour for total head of 50 feet.
5 inch	20	2	3	350
6 "	70	3	4 to 5	1000
8 "	90	4	6 to 7	2000
10 "	100	5	7 to 8	3500

Directions for Running the Reeco-Rider Engine.—The following instructions will be found helpful in the management of this engine.

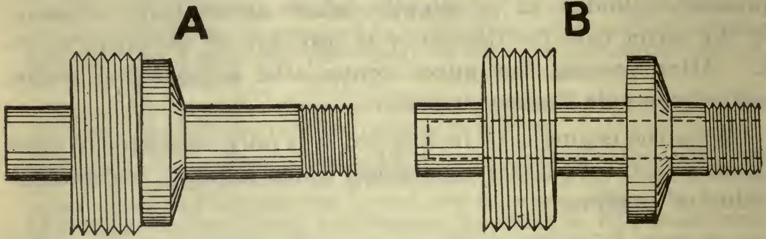
Firing.—If gas be used as a fuel, do not apply the light until the gas has been turned on three seconds. The proper heat is obtained from blue flame, though sometimes it is necessary, where gas is deficient in heat, to have a little red in the flame. If engine run too fast, shut off the gas until the proper speed is obtained.

If coal be used, the fire should be lighted one-half to three-quarters of an hour before it is required to start the engine. Use anthracite coal of chestnut size if it can be obtained. Other kinds of fuel will answer, but this is preferable.

When the engine is hot enough to have its full powers, the heater—the casting hanging over the fire marked "F" in sectional cut—should be a dull cherry red. If this color cannot be obtained, the draught is defective or the fuel deficient in heat. To ascertain whether the heater is red, insert small fire shovel in the fire chamber so as to prevent the reflection of the flame on the heater, and this will allow you to see whether the heater is red or not.

Keep a thin, bright fire when the engine is running. Keep the ash pit clean. Do not allow engine to stand still with a fire in it and the draught wide open. Close the lower and open the top door whenever your engine stops, even for five minutes; and when through pumping, open top door.

If the engine run faster than desired, open the top door a little.



FIGS. 3,276*w* and 3,277.—Gas pipe connection. **A,** position when lighting gas or turning off; **B,** position while gas is burning. ***In setting up*** one of these engines with a gas furnace, it is advisable to have gas pipe sufficiently large, in order to keep the pressure of the gas up to its proper point when it is delivered to the burners, otherwise the burners will not work satisfactorily. The proper size is ½ inch for 5 inch engine; ½ inch for 6 inch engine; ¾ inch for 8 inch engine; and ¾ inch for 10 inch engine. These sizes will answer unless the pipes are long, when they should be increased. Connect gas pipe to threaded end of A. Put a cock in gas pipe, within one or two feet of engine, and regulate supply of gas by opening this cock until the proper flame is obtained. If the directions concerning the position **A** are not obeyed, there will be a report like a small pistol shot, but no damage will be done. If position **B** is not kept when gas has been lighted, not enough heat will be got from the gas. The right heat is obtained from a *blue* flame, and engine should start in from ten to thirty minutes after lighting gas. A little observation of the flame will teach the attendant how to start the engine in the shortest time.

Oiling.—Never use lard oil or any gummy oil. An excellent mixture is a compound of paraffin and sperm oils, in the proportion of two parts of the former to one of the latter. The manufacturer recommends an oil known as "Rider Engine Oil, No. 20."

Oil the engine every time it is used, but apply only just enough to the pistons to keep them always bright. Too much oil is injurious, as well as wasteful.

Never allow the oil cups to get empty. Be sure that the end of the wick passes down the tube to the shaft or pin. A few drops of oil should be put

Cleaning.—*The effect of neglect to keep the interior of the engine clean is to greatly increase the friction, whereby much of the power of the engine is lost, and consequent* **overheating** *must be resorted to to make the engine continue to work effectively. Continuance in this overheating will result in burning out the heater. This will* **never** *happen if the instructions in regard to cleaning be observed.*

When the engine is in need of cleaning, it requires a heavier fire and longer time to get it in operation. It also will be apparent by the difficulty experienced in starting, as the hot piston will show a tendency to stick in the bottom of the cylinder. This may be seen when the engine is **cold** by noting whether the wheel can be turned round freely with the air cock **open.** If not, the engine should be cleaned.

To clean the inside of the engine, disconnect the connecting rods, remove the wheel and leather packings; the pistons may then be drawn out and scraped thoroughly. In scraping the inside of the hot cylinder, be very careful not to allow any of the dirt to get down into the narrow part of the heater. This can be prevented by stuffing waste or rags into the heater, and removing all scrapings with the rags after cleaning the cylinder. It is best to put in two or three layers of rags, making sure they touch the interior surface of the telescope, so that any dirt accidentally falling off in removing the first layer will be caught on the second, and so on.

In replacing the piston, see that it is not turned around.

The Refrigerating Plates.—These must be thoroughly cleaned of all grease. They are contained in the passage between the cylinders. The grease usually accumulates on the "cold" end. The compression piston seldom requires cleaning.

Pump.—There is a small air cock in bottom of pump. If the air barrel should fill with water, it may be recharged with air by opening this cock (with water shut off) for a few moments while engine is in motion. Should the pump leathers or "buckets" swell, it will make the engine labor and sometimes prevent its working. In this case, take out pump rod and trim down buckets with sharp knife until the rod will sink to bottom of pump tube by its own weight. It may be necessary to prime the pump for a few times at first until the valves get thoroughly seated and the leather soft (except when the water flows into the pump from the main or other head). A priming cup is attached to top of pump for this purpose on engines pumping from wells or other places where water does not flow into the pump.

Before starting engine the first time, the valves of pump should always be taken out and scraped clean, as, after being tested at the Works, the valves sometimes "seat" too tight. If this be not done, it may be impossible to draw water. Should a deep well pump ever fail to deliver the water, unless the buckets are worn out, it will be on account of this.

Starting.—Standing at the fire door, pull down on the fly wheel till the hot piston is raised a few inches; then **reverse** the motion till the spring of the air offers resistance, when the forward motion can be easily made. When the pistons balance at the upper part of their stroke, they will come down without help till the hot piston is nearly at the bottom.

A *little* air being now let out, by opening the cock at the bottom of cold cylinder, the crank on the hot end will stand vertically downwards. Shut the cock and pull the wheel past this point of the greatest compression, and the engine will start off at once if sufficiently hot. This is not necessary in starting the small sizes.

Stopping.—To stop the engine, open brass cock in end of bed plate under compression cylinder.

Heater.—If the heater should be burned (or cracked), it becomes apparent through the air blowing through down on the fire.

The heater should be renewed at once, for if overfiring be persisted in, the "telescope" will be warped. ***To replace the heater,*** open the fire jacket and take off. Then remove heater, which is held in place by bolts (marked F, on sectional cut) and make joint for new heater of asbestos and pasty red lead. Screw bolts up tight, and do not put fire into engine for five hours. The telescope is held in place by screws. In putting back these screws, be particular in observing that no air leaks past them.

Check Valve.—The supply or check valve projects from the back of the compression cylinder, and must always be kept in a vertical position with opening at bottom.

It must be kept tight and free, in order that the air may be taken in through it to supply any leakage. Should a hissing noise be heard about it, the air is escaping through it. Take apart and clean with clean rag.

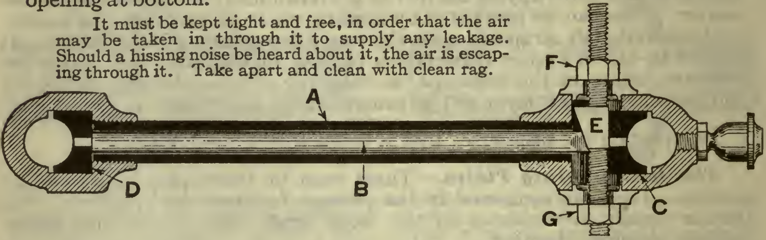


FIG. 3,277*a*.—Reeco-Rider connecting rod. It would be well to take the key E out, so that its action and the construction of the connecting rod may be fully understood. It will be seen that it is wedge shaped, one side acting on the upper brass C, the other on the lower brass D, through the intervention of a rod B, passing through the iron tube A, which connects the two brasses or ends of connecting rod. Replace the key with the straight edge up, and tap lightly till it bears upon both its edges. Use no force in this operation. Next, screw up the nut G on the small end of the key till it bears on the washer. This can probably be done with the fingers. Then screw up the nut F on the broad end with a wrench, which will slightly withdraw the key. Be particular to always adjust the keys in this manner, otherwise they will be too tight. Should it become necessary to remove the connecting rods from the pistons (to adjust the brasses), it can be done by unscrewing the tap bolts, which hold the knuckle to the piston. A long socket wrench is furnished with each engine for this purpose.

Waste Pipe on Hot Cylinder.—The use of this small pipe is to keep the piston packings moderately cool.

The water need only be allowed to drip out slowly, so that the packings never approach a boiling temperature, as the leather will not suffer injury below the boiling heat of water. The small cock on the front of the engine regulates this waste.

Caution.—In cold weather always open the cocks in bottom of compression cylinder and in bottom of pump.

If this be not done, the frost may crack those parts. If a hissing noise be heard about the engine, it indicates an escape of air. The leak should be found and stopped immediately, or a larger fire will be required, which will in time damage the heater. This is *vitally important.*

Keep the engine wiped clean. Five minutes a day thus spent will not be wasted time. Avoid any lost motion in the connecting rods, as it is accompanied by both noise and wear of the brasses.

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